THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

VOLUME I.

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CONTENTS

page

AERONAUTICS:

First Training of Parachute Troops 244
"The Engineers' Digest," Vol. 1, No. 7, p. 244.

Plastic Planes 231
"The Engineers' Digest," Vol. 1, No. 7, p. 231.

Development of the Technique of Riveting Light Metal 242
"The Engineers' Digest," Vol. 1, No. 7, p. 242.

ELECTRICAL ENGINEERING:

A Gap in the Protection of Buildings against Lightning: Protection against Endangering by Lightning owing to Overhead Network 249
"The Engineers' Digest," Vol. 1, No. 7, p. 249.

Diesel-Electric Locomotives 235
"The Engineers' Digest," Vol. 1, No. 7, p. 235.

Unipolar Generator for Low Voltages and High Currents 261
"The Engineers' Digest," Vol. 1, No. 7, p. 261.

MECHANICAL ENGINEERING:

A Contribution to the Question of Increasing the Efficiency of Machine Workshops 233
"The Engineers' Digest," Vol. 1, No. 7, p. 233.

Combustion of Waste Wood Products 254
"The Engineers' Digest," Vol. 1, No. 7, p. 254.

German Machine Tool Accessories 252
"The Engineers' Digest," Vol. 1, No. 7, p. 252.

New Swiss Machine Tools 237
"The Engineers' Digest," Vol. 1, No. 7, p. 237.

HEATING AND VENTILATING:

The Thermo-Pump Heating Plant of the Renovated Town Hall at Zurich 247
"The Engineers' Digest," Vol. 1, No. 7, p. 247.

METALLURGY:

Molybdenum in Iron and Steel 256
"The Engineers' Digest," Vol. 1, No. 7, p. 256.

The Treatment of Carbonyl-Nickel Powder in Melting Processes in the German Iron and Steel Industry 260
"The Engineers' Digest," Vol. 1, No. 7, p. 260.

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PRINCIPAL CONTENTS

Improvements in Electric Railways.

A Two Pole Alternator of 100.000 KW built by the Factory "Electrolisa," Leningrad.

Strengthening of Gas Welded Seam by Recrystallisation.

Internal Combustion Engines for Special Fuels.

The "Gilli" Locomotive.

High Speed Rotary Pumps.

Chromium Plating of Machine Parts.

Comparative Performances of Engines using Carburettors.

A New Gas Turbine Power Plant.

Improvements in "Benson" Boilers.

Electric Catapult for Launching Aeroplanes.

Tinder Removing Machinery.

Aerodynamic Testing of Turbine Models.

Aero-News.

Patents and Inventions.

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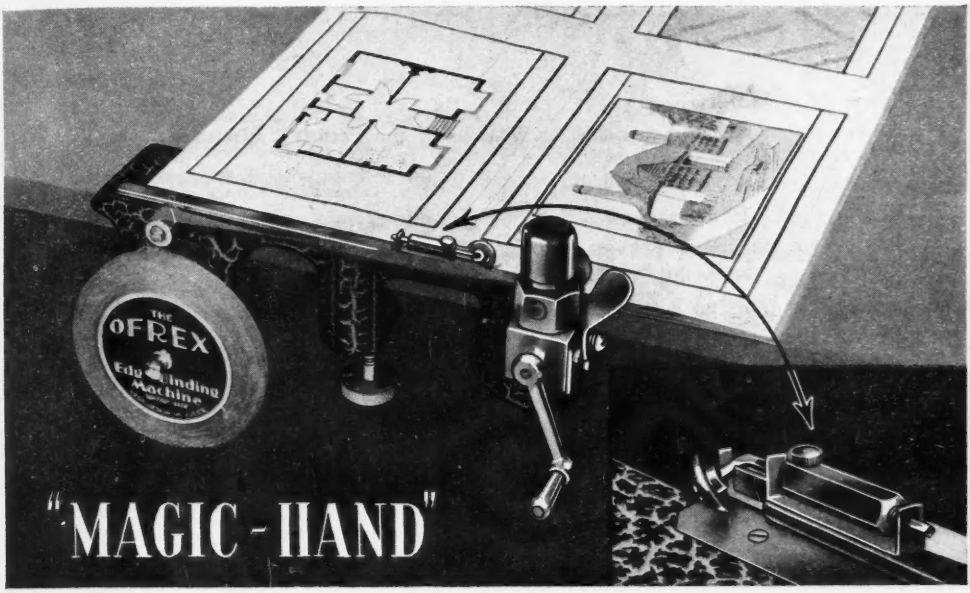

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CONTENTS

	page
Internal Combustion Engines for Special Fuels.	4
A Two-pole Alternator of 100.000 KW Built by the Factory "Electrolisa," Leningrad.	6
Improvements in Electric Railways.	11
Comparative Performances of Engines using Carburettors.	14
Electric Catapult for Launching Aeroplanes.	18
Aerodynamic Testing of Turbine Models.	19
A New Gas Turbine Power Plant.	23
Strengthening of Gas Welded Seam by Recrystallisation.	26
Corrosion of Aluminium by Electric Currents.	31
Autogene Surface Hardening.	32
Preventive Measures to Protect Air-Heaters against Corrosion.	33
Female Labour for Welding.	35
Surface Hardening with Coal Gas.	36
New Possibilities in Heating Smelting Furnaces.	37
Oil Glands of Bearings.	38
Preventing Damage in Boilers.	38
Cracking of Tubes owing to Oscillation.	39
The "Gilli" Locomotive.	39
Vierendel Girders in the Ilva Steel Works.	40
Tinder Removing Machinery.	42
Small Iron-Clad Rectifiers.	43
Method of Examination of the Relative Swinging of Overhead Lines. ..	45
Photoelectric Control in Transport and Supply.	45
Chromium Plating of Machine Parts.	46
The Stress in Openings of Boiler Shells.	47
Improvements in "Benson" Boilers.	48
Utilising the Heat of the Earth.	49
Experimental Determination of Fluctuating Heat Flow.	50
Miniature Ball Bearings.	51
Model Tests of a Wind Power Station.	52
The Design of Electrodes for Spot Welding of Iron and Steel Plates. ..	53
Aero-News.	54
Patents and Inventions.	55
High Speed Rotary Pumps.	56

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ASKING FOR CRITICISM

The phenomenal progress in almost every branch of applied engineering, accelerated by the needs of warfare make it difficult, if not impossible, to keep abreast with the latest improvements.

The importance of sources mediating developments to the British industry is therefore evident. The aim of this journal is to submit articles selected from foreign technical periodicals which are in close and apparent relation to actual engineering problems, to the British engineering world.

" The Engineers' Digest," devoted exclusively to the publication of articles on engineering progress abroad, will cover a field which has not yet been sufficiently represented in the excellent technical press of this country.

The helpful understanding of the Trading with the Enemy Branch of the Board of Trade made it possible to give account not only of the improvements in neutral countries but to take in and epitomise the current technical papers of the enemy and of enemy occupied countries.

Presenting the first issue of " The Engineers' Digest," I should like to ask for the severest criticism of its contents. I shall be grateful for critical hints which I shall consider as assistance in avoiding mistakes that may have been made in the first issue, and in finding out which way the evolution of this journal is to take in order to render a real service to the British engineering world.

Suggestions as to the following problems would be especially appreciated : Introduction of the U.D.C. Indexing system in later issues ; Introduction of a systematical arrangement of the articles, a feature which has been intentionally omitted in this first number ; Length of the abstracts (whether they are sufficiently long for thorough understanding and practical use) ; Hints in regard to the contents, etc.

I am pleased to express my sincere thanks to Mr. J. E. Montgomrey, B.Sc. (Eng.), M.I.Mech.E., Secretary of the Institution of Mechanical Engineers, and to the Members of the Advisory Committee for their moral support and efficient assistance in paving the way for this new enterprise.

J. EMERICH PAJZS.

INTERNAL COMBUSTION ENGINES FOR SPECIAL FUELS.

By Dr. Ing. F. Dreyhaupt. (Condensed from *Motortechnische Zeitschrift*).

Coal Tar Oil.

Experiments carried out at the Research Institute in Dresden showed that Diesel engines may very well be fuelled with coal tar oils, provided the temperature of the air in the combustion space is sufficiently high.

Fig. 1 shows the antechamber of an engine with a built-in air heater. This heater has a number of narrow passages through which alternately the hot combustion gases and the air have to pass. Thus, right at the beginning of the compression stroke the air temperature in the antechamber is considerably higher than in the working cylinder. As the compression proceeds this temperature difference becomes still greater, partly owing to the well-known nature of adiabatic compression and partly because also the successively supplied air is steadily heated. With properly designed heater the engine may even run for a considerable time with no load. For the start, however, an electric heater coil was found to be necessary.

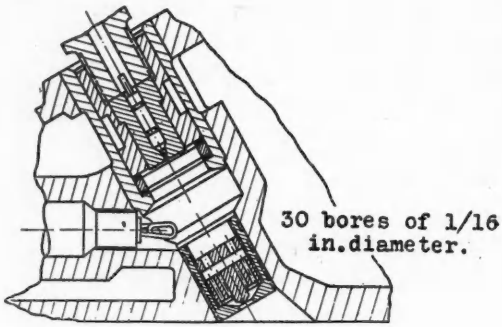


Fig. 1

The test showed further that for full load the heater's effect was particularly favourable, as it is stated to have reduced the ignition lag and the specific fuel consumption to somewhat below the level, which is normal for a usual Diesel engine. These results are said to have been obtained with four-cycle engines of 73 and 116 cu. in. stroke volume, 1200 r.p.m., compression ratio 17, 5, resp. 16. Subsequent experiments with a two-cycle engine of 116 cu. in. stroke volume are claimed to have been successful, even at a compression ratio of 13, 5 and 650 r.p.m.

Powdered Coal.

For the last 40 years experiments were carried out to fuel engines directly with powdered coal, the different powdered coals having an ignition temperature of 300 to 600 deg. C. It was found further that the coal powder may contain also some bigger sized granules without any detrimental effect. Regarding the engine's speed at the present the maximum is stated to be about 400 r.p.m.

Fig. 2 shows an engine designed by *Pawlikowski* in 1928. The engine is fitted with a sluiced antechamber **b**, into which the powdered coal is admitted through valve **a** at about the end of the suction stroke. Valve **a** is equipped with a de-aerating valve and passage, through which the pressure in **b** is apparently relieved throughout the admission period. The powdered coal is supplied by two endless screws **d**, and valve shutter **g** seems to regulate the quantity to be admitted, which is apparently blown in by low pressure air supplied through **f**; the use of low pressure air being ensured by de-aerating valve in **a**.

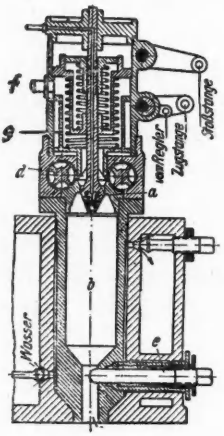


Fig. 2

The combustion in **b** is started by self-ignition, which may be governed by throttling with valve **e** the air passage **c** to the working cylinder. After a part of the powdered coal has burnt out in **b** the pressure increases in the same, and the rest of the coal is blown into the working cylinder, where the rest of the combustion will take place.

The operation of the afore-described engine was not found to be very satisfactory, particularly because the supply by endless screws was not too reliable when the coal powder contained some moisture. Similar difficulties arose, too, regarding the valve shutter **g**.

With a view to the above difficulties a new process of operation has been designed by *Nägel* and *Zinner* in 1934 at the Research Institute in Dresden. For this engine the pressure difference between the sluiced antechamber and the working cylinder is being utilized. Fig. 3 shows the engine, and Fig. 4

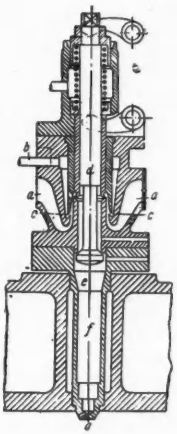


Fig. 3

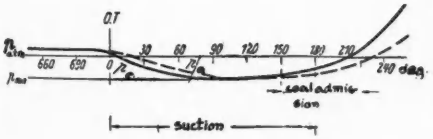


Fig. 4

Pa = pressure in antechamber.
Pc = pressure in working cylinder.

the pressures during the suction stroke in the antechamber and in the working cylinder. Owing to the inertia of the air, for the first part of the suction stroke, the pressure in the working cylinder is less than in the antechamber; this phenomenon reverses, however, for the second part of the compression stroke, while all the time both pressures are below the atmospheric one. Thus during this second part of the compression stroke valve **d** opens, and the powdered coal accumulated in **a**, is sucked into antechamber **f**, and cannot get into the working cylinder because the pressure in the latter is higher than in **f**, though still below the atmospheric one. During this suction period the powdered coal is further mixed at **c** with atmospheric air incoming through **b**. The throttling of the additional air, let in through **b**, provides also a simple means to govern the engine, and is further supported to this end by a cylindrical shutter. It may be yet added that the antechamber's volume amounts to some 15% of the total combustion space.

Fig. 5 shows the development of admission valve **d**, and the best results are stated to have been achieved with design **C**.

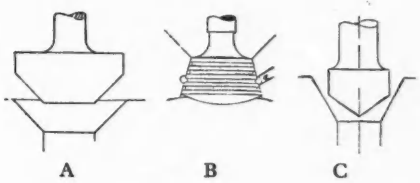


Fig. 5

As a final compression pressure 440 lbs. sq. in. is suggested for the above engines because the evaporation of moisture and the freeing of different gases, absorbed by the powdered coal, is claimed to reduce the temperature by about 180 deg. F.

The mechanical efficiency of a 160 BHP engine is stated to be 70% and the effective fuel consumption to be 9920 BTU/BHPe, corresponding an efficiency of about 25%.

The engine's wear in the cylinders is claimed to have been lately reduced by employing special high grade materials, and the cylinder lubricating system is entirely separated from the rest of the engine.

A TWO POLE TURBO-ALTERNATOR OF 100,000 KW BUILT BY THE FACTORY "ELECTROLISA," LENINGRAD, U.S.S.R.

By M. L. BRITSYN. Condensed from *Vestnik Electro-Promishlennosti, Moscow.*

THE Factory "Electrolisa" has completed a two pole turbo-alternator of 100,000 KW, 3000 r.p.m., and the same has been in service for a year at the Stalinogorsk Power Station. This is stated to be the biggest unit ever built so far, and Table I. shows its main characteristics.

Table I.

Temperature of the entering air. Deg. F.	TURBO-ALTERNATOR							
	OUTPUT		Voltage	Current in Stator	Cos θ	R.P.M.	Cycles/sec.	Current in Rotor. Amp.
	KVA	KW	V	A				
104	111,100	10000	15750±5%	4075±5%	0.9	3000	50	560
86	117,500	10000	17750±5%	4300±5%	0.85	3000	50	600

	EXCITERS				
	Output KW	Voltage V	Current A	R.P.M.	Excitation
Main exciter	220	350/600	630	3000	Independent
Pilot exciter	2/6.5	115	17.4/56.5	3000	Compound

The rotor's circumferential speed is 509 ft. per sec. at 3000 r.p.m. and 610 ft. per sec. at 3600 r.p.m. after starting. The force acting on one tooth and trying to tear it from the rotor amounts to 2500 t. This is 6300 times the weight of the piece which has to withstand the stress. The forging of the rotor has been carried out at the factory "Barricade" and table 2 shows the quality of the material used. Fig. 2 shows the alternator, the main exciter and pilot exciter. The winding of the rotor went through a baking process, the insulation belonging to classification 3, for which the permissible maximum temperature is 266 deg. F. The connections of the winding are covered with aluminium covers and stiffened by aluminium wedges. As protection against extension of the winding, the connections are resting against springs.

The rotor-end-bells are particularly strengthened by being first driven on a ring and then shrinking this ring on the rotor and a nave which is being fixed to the rotor by bolts.

The slip-rings are fixed on the same side as the exciters immediately behind the main bearing. They too are first shrunk on micanite-insulated rings.

The connection between winding and slip-rings pass through the central hole of the rotor. The connection with the winding and the slip-rings is carried out by special bolts and gudgeons. The surface of the rotor is covered with small grooves so as to increase its delivery of heat.

Table 2.

	Actual Stress at 3600 r.p.m. in tons per sq. in.	PROPERTIES OF MATERIAL			Reserve of Strength compared with strength at yielding point	Material
		Ultimate Strength in tons per sq. in.	Yield Point, tons per sq. in.	Max. Elongation to a test bar of l=5d		
Tooth of Rotor	17.8	45.7	29.8	15%	1.68	Chrome-Nickel-Molybdenium Steel
Central opening of Rotor	18.05	45.7	29.8	15%	1.655	
Rotor-end-bells	30.95	63.5	55.3	28%	1.79	Non-magnetic "Anstenit" Steel

In Fig. 1 the proportions of the machine can be very well seen. The slots, e.g., had to be milled on a special milling machine with two saddles.

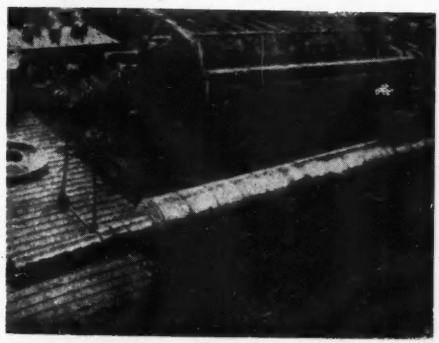


Fig. 1

Fig. 2 illustrates how the winding is laid into the slots, the upper turns of the last coil just being fixed. Considering the length of the rotor it was necessary to prepare special trestles to support the coils when these were being laid into the slots. While winding was going on, three winding specialists were working on each side of the rotor.

The whole stator is completely welded and is provided with several channels for ventilation. The size of the stator was limited by the capacity of the factory's cranes and by rail transport conditions. The weight of the active iron made it necessary to fit in the winding at the place of destination—the Stalinogorsk Power Station.

Considering the length of the machine and the necessity of using fans for cooling, the air conduct had to be solved in an unusual way. In Fig. 3 the preparations are shown for the fitting of the stampings. On the same Fig. the six outlets for the hot cooling air can be seen. Under the paws in the inclined part of the machine five inlet holes for the air are situated on each side. The weight of the stator—about 160 t—makes the change from vertical to horizontal position very difficult and a special mechanism had to be devised for this purpose. The active iron parts are composed

Fig. 2

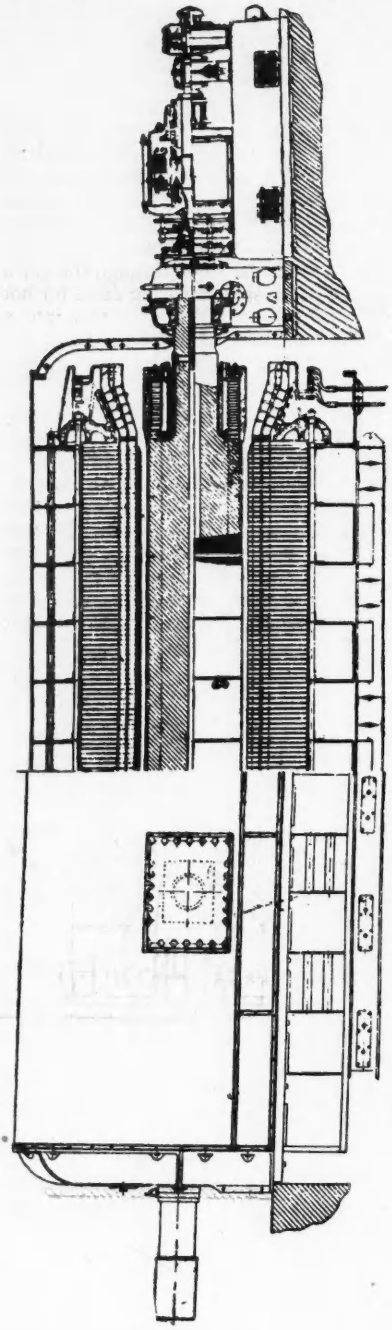


Fig. 4

of alloyed high quality steel stampings covered with special insulating varnish. The stampings are held in place by retaining bolts and at the ends by plates of non-magnetic cast iron.

The stator has a bar winding consisting of two layers with continuous micanite insulation on asphaltum lac. The bars are built up of twisted square wires, the winding having a shortened pitch. This arrangement is stated to provide small additional short circuit losses and saves also copper. Several layers of micanite with a long compounding treatment in vacuum is said to ensure good dielectric strength. The whole winding is wrapped with a semi-conducting asbestos tape, which is claimed to contribute with a varnish of the same property to compensate the potential difference along the bars, and to improve at the

Fig. 3

same time the dielectric strength of the insulation. The voltage of 15750 V is claimed to be the highest ever applied in machines of the factory. The bars of the winding are said to be the longest and their cross section the biggest ever produced. Nevertheless the winding is stated to have been completed within a very short period and the percentage of refuse to have been very small. The maximum permissible temperature of the winding is 212 deg. F. Thus the temperature rise is 140 deg. F. if the entering air has a temperature of 104 deg. F, and 158 deg. F if the air is only of 86 deg. F.

The body of the stator is covered on both ends with cast iron shields. The shields have on each side two glass covered openings which make it possible to watch the winding constantly.

A steam pipe has been fixed all along the stator and also in its frontal parts. The pipe is provided with a great number of small holes for the ejection of steam in case of some damage. Under the stator near the outlets for hot air (Fig. 4) a device can be seen to stop the cooling air in case of breakdown. The airflux is governed by remote control from the main panel.

The ventilating plant (Fig. 5). Ventilation is provided for by fans. The holes for the inlet air are distributed along the sides of the alternator. From the inlets the air is streaming to the surface of the active iron and through ventilating ducts to the gap between stator and rotor and from here further on to the chambers at both ends of the machine, cooling first the connections of the stator winding, then those of the rotor winding and leaving finally through holes in the big teeth of the rotor. The air is also moving along the rotor through the gaps, and leaves through six ventilating ducts for hot air. From these ducts the air enters into a chamber for hot air situated under the alternator. The chamber contains four fans which are put into the stream of hot air. This is claimed to have an advantage combined with the position of the fans in the stream of cold air, because it prevents the cooling air from being heated while passing through the ventilator. In the wall of the chamber for hot air an opening has been left covered by a filter which establishes a connection between the chamber and the outside atmosphere. Additional sucking of air into the ventilating system is possible only through this filter. In the

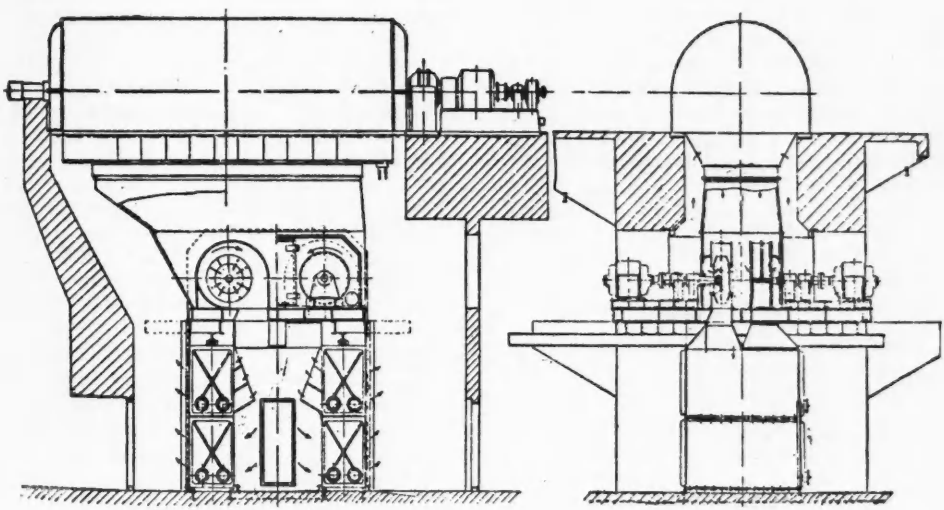


Fig. 5

other parts of the ventilating system there will be always a higher pressure than outside, thus preventing the outside air from entering and permitting only the air to escape through leakages. The chamber to which the cooled air is sucked is covered with heat insulation so that the cooled air should not receive any heat on its way to the turbo-alternator.

Just behind the junction of the air ducts from the fans with the lower chamber there is under each fan a valve for the air. The valve's own weight and the ventilating pressure keep the same open but a pressure difference closes them automatically if the fan were not in action. If all four fans are out of action the valves open under the influence of their own weight. A uniform distribution of air in all parts of the turbo-alternator is claimed for practically any combination of the fans. The results of tests about the air consumption with different fans in action are shown in Table 3.

Table 3.

Number of fans in action	0	1	2	3	4
Consumption of air in cu. ft. per sec.	0	1060	1835	2400	3080

In normal service only three fans are working, the fourth being kept in reserve. The power needed by each fan amounts on the average to about 100 KW.

The weight of the turbo-alternator per I KW is claimed to be 4.84 lbs. Thermodetectors are distributed over the whole of the machine enabling a constant watch.

Electrical characteristics: The winding of the stator has two parallel branches per phase, star connection being chosen. In order to ensure longitudinal and transversal protection of the winding it has been provided with twelve ends. The application of a pilot exciter enables a rapid change of the voltage of the rotor of the main exciter to take place and consequently also the voltage on the slip-rings of the turbo-alternator.

According to oscillograms the speed of the voltage increase is 600 V/sec or 170% of the exciter's normal tension. The voltage regulation of the main exciter is executed by a wheetstone bridge type regulator. The main exciter has, beside its normal winding, a supplementary one which is fed through a special regulator. This system is stated to increase the dynamical stability of the alternator.

The electric and thermal tests of the alternator are said to have fulfilled all specifications of the U.S.S.R. standards concerning electric machines. Furthermore the results of electric tests are said to have surpassed always the calculated ones.

In Table 4 are compared the results of experiments and calculation of the losses (with exception of the mechanical losses on which there are so far no experimental results).

Table 4.

Losses	Calculation KW	Tests KW	Reduction KW
Iron and additional no-load losses	485	460	25
Ohmic losses in stator winding	108	108	—
Additional short circuit losses	184	172	12
Ohmic losses in rotor winding	180	160	20
Sum.	957	900	57

Table 4 shows that the losses at normal load of 100000 KW and 0.9 power-factor amount to 0.9%. The above shown test results allow us to expect a very good total efficiency when mechanical and other losses will be included.

IMPROVEMENTS IN ELECTRIC RAILWAYS.

By Dr. Ing. K. Sachs. (Condensed from *Schweizerische Bauzeitung*.)

The competition of road and air transport enforced the improvement of railways, especially as regards comfort and speed. As a first step it was thought to reduce the weight, as the tare weight per person is on the average for a car 5 cwts. and for a motor cycle 1 cwt. At the same time the tare weight per person averages for a quick train about 1½ t., for a slow train 1 t., and for a train with sleepers about 5 t. per berth. Lately it has been possible to reduce the weight, and the double railcar of the S.N.C.F. for Region de l'Ouest weighs, for instance, only about 8 cwts. per person. The extensive employment of welding effected a saving of 15 to 20% in weight. By using high grade steel of 44½ t/sq. in. tensile strength, as compared with the usual 32 t/sq. in., further 20% could be saved, especially if hollow axles were used instead of full ones. For example, a hollow axle with 6 2/3 in. outer and 5 in. inner diameter, while having the same moment of resistance as a full axle of about 5⅞ in. diameter, is actually some 58% lighter. It might be mentioned, too, that Messrs. *Oberschläsische Hüttenwerke A.G., Gleiwitz* manufacture hollow axles by welding two halves together. For small numbers however the cheapest way was found to be to drill each axle separately.

Regarding carriage wheels, a number of different improvements have been carried out recently. First of all, through the lighter body weights and thus smaller axle pressure, as well as by using higher grade steels for the wheels too, the dimensions of the wheels' rims and hubs could be reduced by 30 resp. 15%, saving some 25% in weight. Messrs. *Bohumer Hüttenverein*, are even said to save 50% weight by building the carriage wheels' discs corrugated in the axial and radial direction.

Fig. 1 shows a tram's rubber cushioned driving wheel. At a wheel pressure of 2 to 2½ t. these wheels are said to last for some 62,500 miles.

Fig. 2 shows the *Michelin* full pneumatic wheel; the wheel pressure is about 1,2 tons and the wheels are said to last for about 21,800 miles. These wheels are stated to eliminate any kind of noise, and through the greater adhesion between steel and rubber the permissible acceleration is about 3,3 yard/sec².

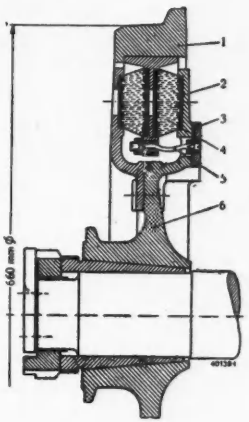


Fig. 1.

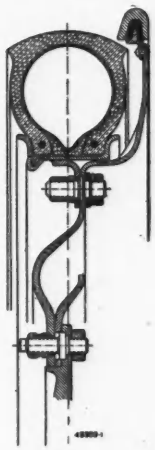


Fig. 2.

1. Bandage; 2. Slide Protection; 3. Slip Ring for earthing; 4. Rubber Pad; 5. Connecting Cable to earth; 6. Wheel Hub.

Fig. 3 shows a new design of truck frame, developed by Messrs. *Henschel & Son, Ltd., Kassel*. In the horizontal section rectangular wheel frame is being kept in position through guiding cranks **2** and **3**, which are normally in the central longitudinal plane of the locomotive. Both cranks are fixed on shaft **1**, and the axis of shaft **1** meets the connecting line of the two guiding points in **P**, this being the motion centre for the corresponding wheels. The oblique shaft **1** is mounted in the locomotive frame. The driving and the trailing axles are connected and co-ordinated through lever **4**, which lies in the plane formed by the two axles.

For locomotives, different elastic spring clutches were found to be superior to joint mechanism, as the latter are subjected to considerable wear resulting in an expensive up-

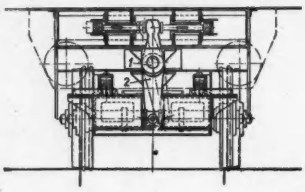


Fig. 3.

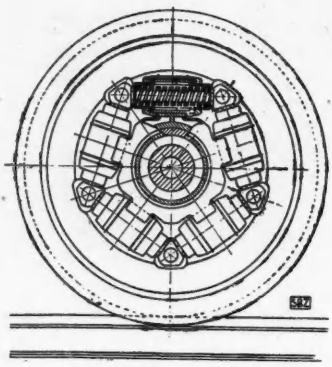
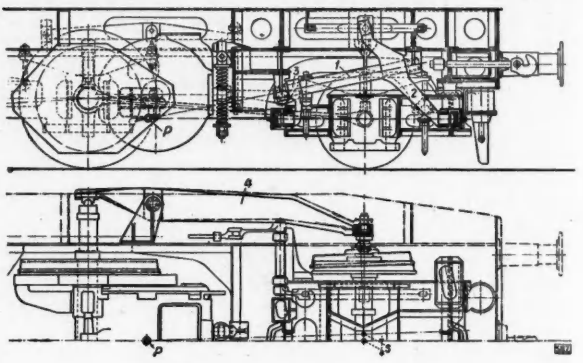


Fig. 4.

keep. Figs. 4 and 5 show the clutches employed for the triple railcar train of the Federal Swiss Railways (10½ cwts. per seat). The driving wheels have a diameter of 35 7/16 in. and the springs work upon radial grippers fixed on the wheels' hubs. This arrangement has the advantage that normal full disc wheels may be used and the entire mechanism is automatically lubricated.

Figs. 6 and 7 show the arrangement by Messrs. *S. A. des Ateliers de Secheron*, as used for different Swiss Railways. The springs work through intermediate spherical tappets upon radial grippers, fixed directly on the axles, or upon suitably formed spokes. As the tappets' ends are ball-like and thus no sliding may occur, it is said that lubrication of these parts is not necessary.

Fig. 8 shows the arrangement as developed by Messrs. *Brown, Boveri & Cie.* The cased tappets are built into the big gear, which is fixed on a hollow shaft, the arms of a gripper fixed on the wheel axle transmitting the drive. The advantage of this arrangement is said to consist in the clutches being inside the gear



lubrication system, and that the torque transmission takes place in the gear's plane.

The Italian system by *C. Bianchi* employing plate springs is shown in Fig. 9 for wheels of small diameter; the driving gear is fitted with jaws grasping the plate springs.

Fig. 10 shows the disc clutch employed by Messrs. *Brown, Boveri*, two tappet discs in connection with two hollow axles allowing deviation in any direction. The torque is transmitted to the outer hollow shaft, which

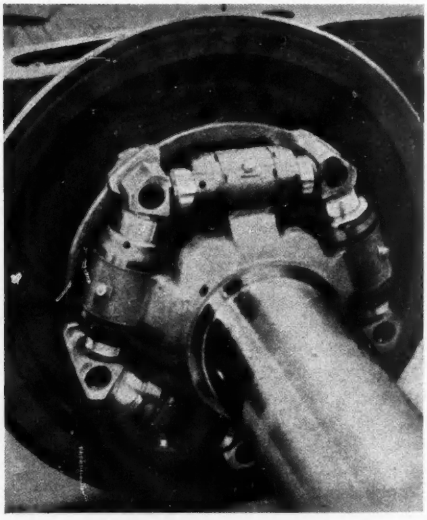


Fig. 5

is connected at one end with one of the two tappet discs both fixed on the inner shaft, whereas the other tappet disc is being coupled with the driving wheel. Thus no part subject to wear is employed.

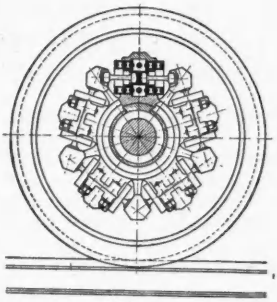


Fig. 6.

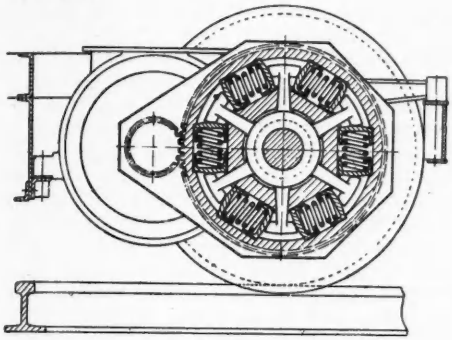


Fig. 8.

The increased speed of modern railcars and trains is 80 to 115 m.p.h. and necessitates a very accurate regulation of the braking. Messrs. *Westinghouse El. & Mfg. Co., Ltd.*, employ for example a pendulum, working upon an electro-pneumatic valve of the brake system,

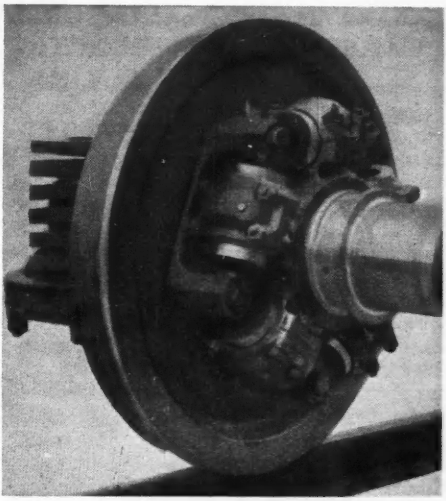


Fig. 7.

the deflection of the pendulum being in the direction of the acceleration. When the train slows down to about 5 m.p.h. this arrangement ceases to operate and the pressure air brake alone stops the train.

Other trains are fitted with mechano-electrical brake systems. The 8-wheeled railcars of the S.N.C.F. are fitted, for example, with a system whereby at first the mechanical brakes take over some 20% of the braking with two subsequent electrical stages. At and after the fourth stage the electrical braking is switched off and the mechanical brakes work to the full extent.

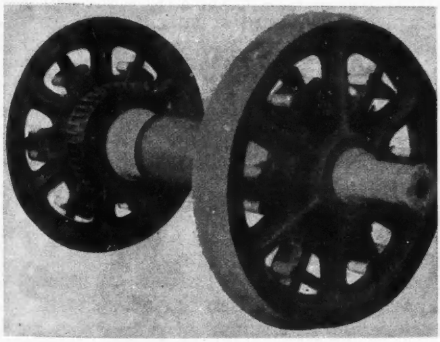


Fig. 9.

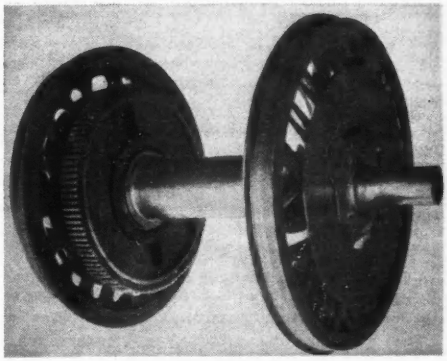


Fig. 10.

COMPARATIVE PERFORMANCES OF ENGINES USING CARBURETTORS, INJECTION INTO THE INDUCTION PIPE AND INJECTION INTO THE CYLINDER

(Condensed from a report of the Langley Memorial Aeronautical Laboratory, U.S.A.)

COMPARATIVE tests using the three systems of fuel supply described in the title were carried out on an air-cooled Wright 1820-G engine, using fuel-air ratios varying from 0.10 to the limit of stability and engine speeds from 1,500 to 1,900 r.p.m. The comparative performances with a fuel-air ratio of 0.08 were investigated at speeds of 1,300 to 1,900 r.p.m. The results show that the power obtained, using any of these methods of fuel supply follows, very closely the volumetric efficiency, the greatest power being obtained by injection into the cylinder. The minimum values for specific consumption of fuel were the same by any of the methods, as were also the cylinder temperatures in any given engine cooled in any given way. The engine used for the test was a four-stroke design of $6\frac{1}{8}$ in. bore and 7 in. stroke having a compression ratio of 7.4, the cylinder head being modified where necessary by the insertion of an inlet valve. The carburettor used was a Stromberg NAL-5, modified by the fitting of needle valves in the main jets in order to regulate the flow of fuel, and when not using the carburettor the latter was by-passed by a straight tube, containing a single butterfly valve, for starting up the engine, in order to avoid as far as possible penalising the performance of the injection methods on account of the loss of pressure in the carburettor. A Compur fuel pump was worked off the engine crank shaft through a reducing gear whereby the injection could be regulated, and the period of injection was varied from 65 to 100 degrees of crank angle according to the flow of fuel, the speed of the pump and the discharge pressure. The cylinder temperatures were measured at 34 points by means of thermocouples and a potentiometer. The pressures in the tank and at the point of admission were measured with hydraulic manometers.

Earlier experiments, using the method of injection into the cylinder, had indicated that in the case of an engine with four valves the best performance was obtained if the injection valve was situated between the two exhaust valves, so as to inject the fuel across the cylinder and in the same direction as the air supply. It had also been found that very good results were obtainable with a single valve situated at the centre of the cylinder head. In the present tests only two positions for the injection valve were adopted : namely, in the rear of the opening made for the sparking plug, and between this and the admission opening. Several different sizes and arrangements of injection orifices were tried, of which the most suitable was found to be a fan shaped slit directing the jet horizontally across the combustion space.

The same performance was obtained using an Eclipse, a Compur or a Bosch injection pump, but the fuel consumption was slightly less with the last named. Tests were carried out using pressures at the inlet valve of 1,000, 2,000, and 3,000 lbs. per sq. in. If injection occurs at the beginning of the stroke there is no variation in fuel consumption, but if the injection takes place very late the best results were found to be associated with the heaviest injection pressures. With injection beginning at 60 or 70 degrees of crank angle after the upper dead point, the same maximum power is obtained whatever the injection pressure, and the specific consumption of fuel is only 0.02 lbs. per H.P. hour lower when using the heaviest pressure than when using medium or low pressure. For the purpose of the comparative tests a pressure of 2,000 lbs. at the opening of the valve was adopted, as an excessive amount of leakage past the piston of the pump was found to occur if the pressure were increased to 3,000 lbs. per square in.

Using injection into the inlet tube, preliminary tests had shown that effective pressure at the brake was approximately the same whatever the timing of the injection. In the comparative tests the injection was initiated at 60 degrees of crank angle after the upper dead centre, experiments carried out elsewhere having shewn that under this condition the condensation of fuel on the cylinder wall was reduced to a minimum. No tests were made to ascertain whether a better performance

might be obtainable using pressures at the opening of the valve other than 2,000 lbs. per square in., or with the inlet valve differently situated.

In each of the test runs the ignition was so regulated as to obtain the best effect. The weight of air used for cooling was maintained constant throughout, and the friction of the engine was measured while revolving it at the required speed. The lubricating oil and cooling air were heated so as to ensure a slight excess of oil, with cylinder temperatures of 160 and 250 deg. F. respectively. Power measurements at the brake were corrected to correspond with standard ground temperature and according to the inlet pressure of the engine, on the assumption that the power of the engine varied directly with the pressure and inversely with the square root of the absolute temperature, The indicated H.P. was obtained by adding friction to the brake H.P. The fuel used in all the tests was in accordance with the U.S. Army standard 2-92. of 100 degrees, and, using this, no knocking was audible at any time.

The comparative performances obtained, using the three different methods of mixing the fuel with the air, are shewn plotted in the illustrations, wherein it will be seen that except at low speeds the difference in mean

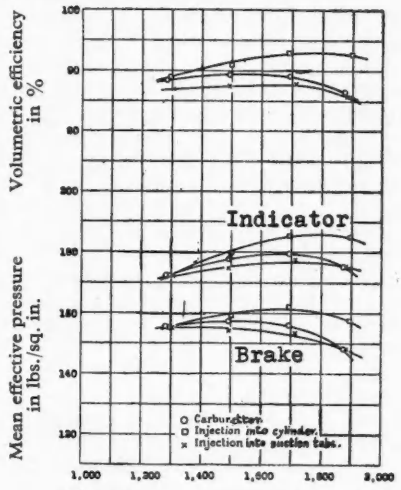


Fig. 1

The engine's output and volumetric efficiency as a function of the speed, for a fuel-air ratio of 0.08.

effective pressure follows very closely the volumetric efficiency. Using injection into the cylinder the volumetric efficiency appreciably increases with speed, whereas using a carburettor this condition is reversed at speeds exceeding 1,600 r.p.m. At 1,900 r.p.m. the volumetric efficiency with fuel injected into the cylinder is 92.5% as against only 86% with the carburettor. The volumetric capacity was unexpectedly found to be lower with the fuel injected into the tubing than with fuel injected into the cylinder, and further test runs carried out at 1,500 r.p.m., using the carburettor, shewed no appreciable difference in volumetric capacity, whether or not the tubing was left in place. Experiments by other investigators have shewn that the lower volumetric capacity obtained when using injection into the admission tube is probably due to an increased volume of the charge attributable to vaporisation within the tube itself, implying a lower temperature (and, therefore, a greater absorption of heat) as the charge passes through the admission orifices and valve, than is the case when using injection into the cylinders. The slightly lower volumetric capacity with injection into the tube as compared with the carburettor is probably due to this cause rather than to the different method of mixing the fuel with the air.

The increased charge obtainable by injecting the fuel into the cylinder is an important advantage, which ought to be utilized to the utmost. It is evidently possible to use tubes of large cross sections, but the dimensions of the admission valves are limited more especially because the velocity of admission must be sufficient to create such turbulence as will ensure thorough mixing of the fuel with the air and assist in propagating the flame. This consideration applies, however, only to slow speed engines or to engines with sleeve distribution depending on a very large admission valve or valves, whereas in modern high speed engines, even if the valves are large, the velocity of admission is ample for the purpose. Using injection into the inlet tube, or a carburettor, the velocity in the tubing needs to be higher so as to ensure mixing of the fuel with the air and prevent separation or condensation. With injection into the tubing this restriction on design is less than with a carburettor, so long as the system is well proportioned. The

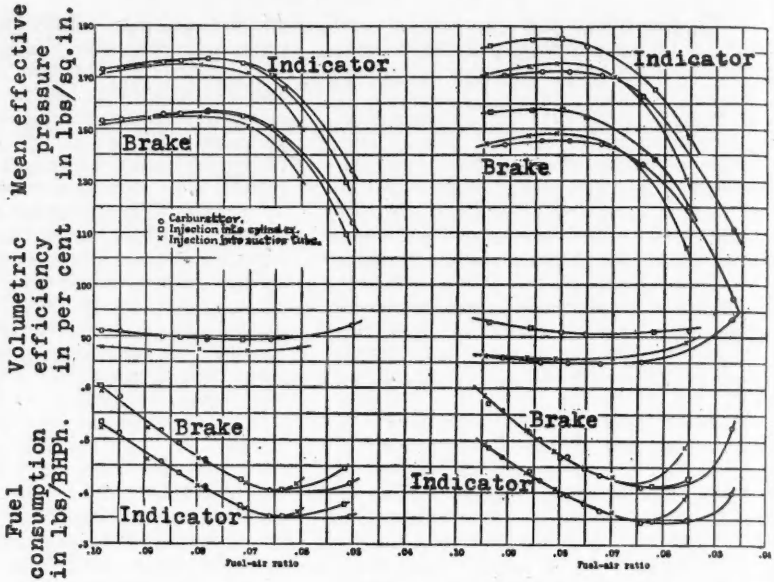


Fig. 2

The engine's output, volumetric efficiency and specific fuel consumption as a function of the fuel-air ratio.

minimum specific fuel consumption is the same for any of the methods of mixing the fuel with the air. When supplied through a carburettor the engine was found to run more uniformly with a weak mixture at all speeds than when either of the other two arrangements for mixing fuel with air were employed. The engine arranged for injection of air into the tubing calls for a richer mixture if uniform running is to be ensured. The variation in strength of mixture required for regular running with the carburettor, or with injection into the cylinder, was lower at high speed than at low speed. A series of equations has been worked out in which the mean temperature at the head and in the bore of the cylinder are expressed as functions of the engine characteristics and of those of the cooling system, and by applying these it is possible to compare the effects of diverse actual cooling conditions.

The upshot of these experiments goes to show that (except for the increased power at high speeds when using injection into the cylinder) practically no difference in performance is to be expected between the three methods of mixing fuel with air, but in practice there are a number of other important factors which enter into the choice. Where a fuel of low volatility is to be used, such as a " safety fuel," the carburettor would not be satisfactory, and the best performance would be obtained by adopting direct injection into the cylinder. Again, where an engine is designed to work with a cut-off in the opening of the valve, it becomes necessary to use injection of the fuel into the cylinder, since otherwise there will be a loss of fuel at the exhaust. Tests carried out on multiple cylinder engines have shewn that by the use of injection better starting is obtained as well as improved acceleration and control. By means of a well designed system of fuel injection it is possible to ensure an almost perfect distribution of the fuel among the cylinders, and at speeds between 2,000 and 3,000 r.p.m., the variation in distribution as between full and part load is less than one per cent.—a figure slightly better than that obtained with a carburettor on a radial engine, and much better than would be obtained using a carburettor on an engine having cylinders in line.

Conclusions.

1. The power efficiency obtained by any of the three named methods of mixing the fuel with the air follows very closely the curve of volumetric efficiency, and reaches its maximum when the fuel is injected into the cylinder. The carburettor offers some resistance to the flow of air, whereas if the fuel is injected into the cylinder very large admission openings can be adopted.

2. For the range of air ratios between 0.10 and approximately 0.06 the same minimum specific consumption of fuel is obtainable by any of the three methods of mixing the air with the fuel. An engine fitted with a carburettor runs more uniformly on a weak mixture, its performance being slightly better than that of an engine with injection into the cylinder and much better than that of one with injection into the tube.

3. For any given power output and cooling conditions the same cylinder temperatures are obtained using any of the three methods of mixing the fuel with the air.

COMPARISON OF CARBURETTOR AND INJECTION

Operation.	*Carburettor.*	*Continuous Injection.*	*Regulated Injection into Induction Pipe.*	*Regulated Injection into Combustion Chamber.*
Distribution.	Depends on Induction system	Depends on Induction system	Excellent.	Excellent.
Atomisation.	Depends on characteristics of jet.	Very good.	Moderate.	Moderate.
Freezing.	May be prevented by special precautions.	Eliminated.	Eliminated.	Eliminated.
Effect of cooling on the fuel charge.	Considerable.	Considerable.	Very small.	Negligible.
Effect of cooling on the charge in the combustion space.	Very small.	Very small.	Measurable in single cylinder unit.	An increase is possible, due to reduction in the temperature of the gas.
Effect on supercharger.	Improves the performance.	Improves the performance.	None.	None.
Scavenging.	Impossible without wastage of fuel.	Impossible without wastage of fuel.	Possible.	Possible.
Fire risk and backfiring.	Eliminated by flame trap.	Eliminated by flame trap.	Eliminated.	Eliminated.
Aerobatics.	Fairly good.	Good.	Good.	Good.
Acceleration of engine.	By special device.	Probably by a special mechanism.	Can be satisfactory without special mechanism.	Can be satisfactory without special mechanism.
Slow running.	Sensitive.	Less difficult.	Good, but special mechanism may be necessary.	Good, but special mechanism may be necessary.
Consumption in cruising.	Depends on distribution.	Depends on distribution.	Reduced by good distribution and probably by stratification.	Reduced by good distribution and probably by stratification.
Maximum power.	Good.	Good.	Favoured by good distribution but hampered by absence of cooling charge.	Favoured by good distribution but hampered by absence of cooling charge.
Simplicity.	No continuous mechanical movement.	Mechanical control usually necessary.	Delicate mechanically controlled mechanism.	Delicate mechanically controlled mechanism.
Regulation.	When correctly adjusted.	Probably delicate.	Carried out by maker of pump but the control would require testing.	Carried out by maker of pump, but the control would require testing.

Operation	*Carburettor*	*Continuous Injection*	*Regulated Injection into Induction Pipe*	*Regulated injection into Combustion Chamber*
Synchronisation.	None.	None.	Fairly sensitive.	Not sensitive.
Weight.	Normal.	May be a little lighter.	Heavier.	Heavier.
Cost.	Normal.	Slightly greater.	Considerably greater.	Considerably greater.
Controls.	Complicated on modern supercharged engines.	Complicated on modern supercharged engines.	More complicated.	More complicated.

ELECTRIC CATAPULT FOR LAUNCHING AEROPLANES

By ING. LEONARDO LO CURTO. (Condensed from *L'Ingegnere*).

AT the exhibition of inventions in the R. Marine pavilion a 1/5 scale model of an electric catapult has been shown. For this catapult the inherent kinetic energy of two rotating steel flywheels is being utilised. The two flywheels are fixed on the axle of an electromotor, which is of relatively small output. There is a drum mounted between the two flywheels on the same axle, and an electro-magnetic clutch couples the drum with the flywheels.

For launching, the flywheels are first brought up to operational speed and then, through the electric clutch, the drum itself is put into operation, winding up the cord attached to the aeroplane carrier. At the end of the run the aeroplane detaches itself automatically to start the flight and the carrier is braked.

The whole operation is controlled by the aeroplane's pilot, and a number of different safety devices are being provided to prevent any mishap.

Fig. 1 shows the catapult and Fig. 2 the results of tests carried out with the same.

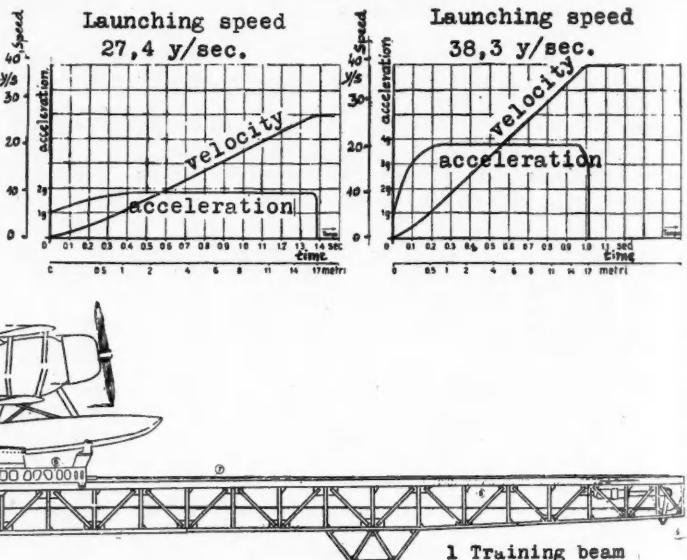


Fig. 2

1 Training beam
2 Bearers
3 Training mechanism
4 Motors
5 Pulleys
6 Carrier
7 Cord
8 Brake

Fig. 1.
Electro-catapult for an aeroplane weighing 5070 lbs.

AERODYNAMIC TESTING OF TURBINE MODELS

By Dr. C. Keller. (Condensed from *Escher Wyss Mitteilungen*).

This latest method of research is not intended to supersede other methods hitherto used, but seeks rather to establish a common basis for carrying out tests and systematic experiments. The present theory of flow phenomena, particularly with regard to modern hydro- and aero-dynamics, when correctly applied to the design of turbo-machinery, seems to reveal surprising prospects for further improvement, especially in the design of high speed machinery.

According to the laws of similarity, under certain conditions air can very well be employed instead of steam, and even instead of water, for carrying out certain tests, the accuracy of the results obtained being sufficient for most technical purposes.

The following abbreviations will be used:

t	=Time
x, y, z,	=Co-ordinates
u, v, w,	=Velocity components
$\rho = \gamma/g$	=Density
p	=Pressure
$\eta = \rho \cdot \nu$	=Viscosity
λ	=Heat conductivity
T	=Absolute temperature
R	=Gas constant
a	=Velocity of sound
A	=Heat equivalent
$k = c_p/c_v$	=Adiabatic exponent
ν	=Kinematic viscosity
Δ	=The operator $\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$

The differential equations illustrating the flow of fluids are:

1. Continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0$$

2. Momentum principle

$$\partial \left(\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial w}{\partial z} \right) = + g_x - \frac{\partial p}{\partial x} + \eta \Delta u + \frac{1}{3} \eta \frac{\partial}{\partial x} \left\{ \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right\}$$

The latter representing the complete equation of motion, including the inertia forces, friction and compressibility. In the case of a gaseous medium the member g_x (inertia forces) may be neglected and the viscosity is assumed to be independent of pressure and temperature.

There are also two more similar equations for the directions y and z:

3. Energy equation

$$\frac{1}{A} \rho g \cdot c_v \left(\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) + p \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) = \frac{1}{A} \lambda \cdot \Delta T - D$$

$$D = \frac{2}{3} \eta \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2 + \eta \left\{ 2 \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial z} \right)^2 \right] + \left[\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z} \right]^2 + \left[\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right]^2 + \left[\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right]^2 \right\}$$

where D (dissipation) is a measure for the friction heat.

4. The gas equation for an ideal gas

$$p = \rho \, g \, R \, T$$

In order to apply the results thus obtained to model tests, all the linear multiplying factors m between the model (index 2) and the original (index 1), i.e., full scale machine, have to be known for temperature, pressure, forces, etc.

$$m_t = \frac{t_2}{t_1}, \quad m\rho = \frac{\rho_2}{\rho_1} \text{ etc.}$$

The four fundamental equations 1. to 4. are first formed for the model and then for the actual working conditions, whereby, for instance, the as yet unknown multiplying factor m is introduced. These equations will then be correct, both for model and the full scale original.

By successively equating the fundamental equations of the model and the original respectively, certain relations between the factors are established, which are necessary for the identity of both cases.

Thus, the equation of continuity 1 reads, when

$\rho_2 = m_\rho \cdot \rho_1$, $t_2 = m_t \cdot \rho_1$, etc.:

$$5.\ \frac{m_\rho}{m_t} \cdot \frac{\partial \rho_1}{\partial t_1} + \frac{m_\rho \cdot m_c}{m_l} \cdot \frac{\partial (\rho_1 u_1)}{\partial x_1} + \frac{m_\rho \cdot m_c}{m_l} \cdot \frac{\partial (\rho_1 v_1)}{\partial y_1} + \frac{m_\rho \cdot m_c}{m_l} \cdot \frac{\partial (\rho_1 w_1)}{\partial z_1} = 0$$

Equation 5. is only identical with equation 1. when the condition of identity:

$$6.\ m_t = \frac{m_l}{m_c} \text{ that is } \frac{t_2}{t_1} = \frac{l_2 \cdot c_2}{l_1 \cdot c_1} \text{ is fufilled}$$

Similarly, the remaining three equations 2.-4. allow of determining the relations for the m-factors from the conditions of identity. These relations provide the laws of similarity which are sought.

Thus the following laws will be obtained in connection with models, which ought to be always applied:

(a) Equal number of atoms for gases, which means:

$$k_1 = k_2 \quad \text{and} \quad k = \frac{c_p}{c_v}$$

(b) Equal Mach's figures: $M_1 = M_2$, where

$$M = \frac{c}{a}$$

(c) Equal Reynold's numbers: $Re_1 = Re_2$, where

$$Re = \frac{c \cdot l}{\nu}$$

(d) Geometric similarity.

In actual practice these conditions can never be entirely fulfilled simultaneously. Theoretical investigations, practical experience and experiments are the factors that have to be taken into consideration when determining to what extent deviations from these laws may be permissible in each single case.

By this new method of research almost all the difficulties connected with experiments carried out with steam as the flowing medium have been overcome. These difficulties were particularly great when experimenting with superheated steam owing to its high temperatures, the small dimensions of the models or parts to be tested, the condensation in the measuring pipes or passages, the high peripheral speeds, the impossibility of accurate observations and the like.

Air may further be substituted for water as long as the velocity of flow of the gaseous fluid is relatively small compared with the corresponding velocity of sound (i.e., when Mach's figure $M = c : a$ is small) the compressibility of the medium being of secondary importance when the changes in temperature and pressure are small.

Suppose now the model of a complete water turbine plant, for instance an up-to-date Kaplan turbine plant, should run with air as the working medium instead of water. The conditions shall be such that all quantitative measurements relating to the model turbine, i.e., efficiency, the quantity of working medium flowing through, determination of the characteristics, etc., should be effected just as in the case of a test plant for water turbines. This operation with air is possible if the values of the admissible velocities of air will be fixed in advance with due consideration to its compressibility.

The equation of continuity 1. for gas flow can be written:

$$7.\ \frac{\partial \rho}{\partial t} + \rho \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) + u \frac{\partial \rho}{\partial x} + v \frac{\partial \rho}{\partial y} + w \frac{\partial \rho}{\partial z} = 0$$

For stationary liquid flow $\frac{\partial \rho}{\partial t} = 0$ and ρ = constant.

Hence:

$$8.\quad \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial t} = 0$$

For plane motion in direction X ($v = o$, $w = o$) the equation of continuity will be:

$$9.\quad f \cdot u = \text{const.}$$

The equation for the flow of water may be deduced from the momentum principle (equation 2). With ρ=const. for stationary, frictionless motion Bernoulli's equation will be obtained.

$$10.\quad p + \frac{\rho}{2} u^2 + \rho g z = 0$$

How the incompressible water can be replaced by the compressible air, may be determined from the relation $\frac{\Delta \rho}{\Delta p}$, i.e. the change of density for a pressure variation Δp. Fig. 1 shows a relation based on the laws of gas dynamics. Fig. 2 shows the error as a function of the velocity of the air, when the dynamic pressure is calculated on the assumption of the incompressibility of air. The correction re-

The influence of the compressibility of the air at various flow velocities on:

The density ρ. The dynamic pressure q. The lift coefficient C

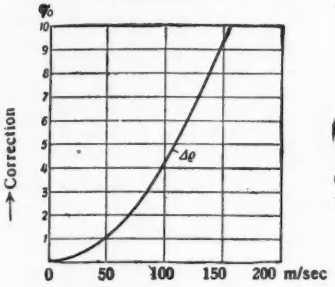


Fig. 1

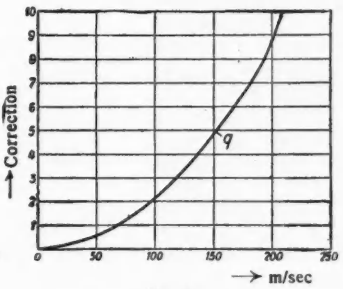


Fig. 2

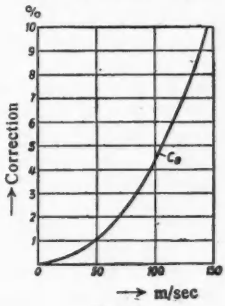


Fig. 3

quired when calculating dynamic forces is shown in Fig. 3 as a function of the lift coefficient c_a.

All these considerations tend to show that, by limiting the maximum velocities of flow (relative velocities at the blades) to about 100 or 150 m/sec., there would be practically no essential difference between the use of air and the use of water, also with regard to quantity. The velocities met with in actual practice are, however, smaller throughout.

The main fundamental principles of such an aero-dynamic investigation plant are shown diagrammatically in Figs. 4 and 5. In the first case (Fig. 4) air is drawn through the system by means of an electrically driven suction fan 1 exhausting into the atmosphere at 2. The air drawn into the system enters through inlet 3 and passes through the guide apparatus into turbine 4, which is being tested, and then flows through the suction pipe into pressure equalizing tank 5 and through a measuring device 6 into fan 1. The inflowing air drives the runner wheel in just the same way as the water would. The power developed by the turbine is measured by a hydraulic or electric brake 7. The actual turbine plant is contained between a and c in the diagram, i.e., it embodies the parts beginning with the inlet and ending with the draught tube. The pressure distribution in the different parts of the system is shown in the lower half of Fig. 4. Since the pressure in the system is below that of the surrounding atmosphere, the entire plant should be absolutely airtight, so as to ensure that only the air actually flowing through the guide apparatus and runner should be measured. As, however, the pressures to be dealt with are small, the vacuum scarcely exceeding 100 mm. water gauge, no difficulties are encountered in the fulfilment of this condition.

As illustrated in Fig. 5 the air may also be forced through the plant. In this case air under pressure is supplied to turbine 4 through

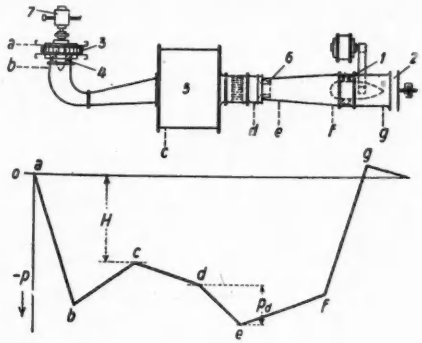


Fig. 4

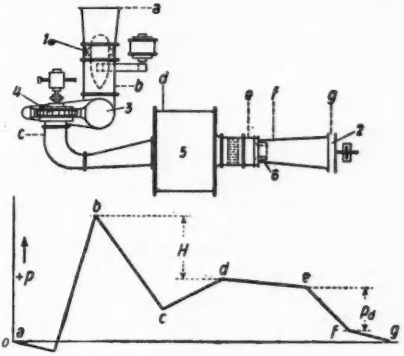


Fig. 5

inlet 3 by a pressure fan ; from the turbine 4 to be tested it passes through the draught tube into pressure equalizing tank 5, and finally through measuring device 6, thereafter being exhausted to the atmosphere at 2. This entire system is under slight pressure ; near the turbine runner, however, the pressure may, under certain conditions, fall somewhat below atmospheric.

The most important condition for obtaining correct results when carrying out such model tests is the close approach to equality of Reynolds' numbers R_e for the model and for the full scale machine. This condition can never be actually fulfilled even in the usual test plants using water as driving medium. In consequence of the small dimensions of the model and the low peripheral speeds as compared with the full-scale machine, the Reynolds' numbers R_{em} of the model is only 1/10 to 1/20th that of the full-scale machine. Experience obtained has shown, however, that this difference in Reynolds' numbers does not affect the final results obtained from model tests to any appreciable extent, and that sufficiently reliable results will be obtained by introducing some correcting co-efficients.

It can be shown too that the Reynolds' numbers are about equal for turbine models of the same size, using air in one case and water in the other. In this case the relation between the Reynolds' numbers is given by :

$$\frac{Re_{air}}{Re_{water}} = \frac{u_{air}\, \nu_{water}}{u_{water}\, \nu_{air}}$$

where u denotes the peripheral speed and ν the kinematic co-efficient of viscosity. At 15 deg. C. and atmospheric pressure : $\nu_{air} = 15 \cdot 10^{-6}$ m²/sec., and $\nu_{water} = 1.15 \cdot 10^{-6}$ m²/sec. With values of $u_{air} = 120$ m/sec. and $u_{water} = 10$m/sec. (this latter figure being the average value usual in hydraulic tests)

$$\frac{Re_{air}}{Re_{water}} = \frac{120 \cdot 1{,}15}{10 \cdot 15} = 0{,}93 \backsimeq 1.$$

with sufficiently close approximation.

From this it follows that the Reynolds' numbers for an aerodynamic investigation do not differ widely from those of a plant using water, as shown by Fig. 6.

The new aerodynamic method of investigation can be applied with equal success to tests on hydraulic pumps. Based on the experience in aerodynamic research a plant working with air was erected in the Works of Escher Wyss some time ago for carrying out tests on pump impellers.

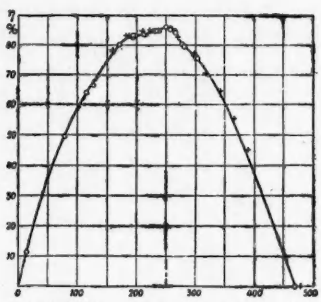


Fig. 6

Reliability of the aerodynamic measurements. Efficiency curves measured on the same model of a Kaplan turbine driven once with air (O) and once with water (+) practically coincide.

The plant was designed to work on the principle of the aerodynamic tunnel with closed or open circuit, and can be thus used for testing models of steam or water turbines. The closed circuit has the advantage that the density of the air may be varied either by admitting compressed air or by drawing air out of the system, so that the Reynolds' numbers may be varied too by changing the density.

In the expression $R_e = \frac{c \cdot l}{\nu}$ the kinematic co-efficient of viscosity ν is inversely proportional to the pressure, assuming constant temperature.

The fact that in such tests the pressure can be varied (this affording a convenient and quick means of also varying the Reynolds' numbers) is of the great importance both for the investigations with steam or gas as with water. Such a variation constitutes the connecting link between the model test and the full-scale machine since, by carrying out tests with different Reynolds' numbers, the correcting co-efficients can be determined without having to carry out tests on models and full-scale machines of different sizes. It is just the compressibility of the air, which at first sight appears to be an obstacle to using air in connection with model tests for hydraulic machines that opens new possibilities in research work, possibilities that cannot be realized when using the incompressible water as driving medium. The first condition to be fulfilled implies, however, that the

tests with air are carried out in the correct way.

From the aforesaid it may be seen that in many cases it is easier to obtain the required Reynolds' numbers in aerodynamic tests than in tests with steam or water. A further advantage offered by the aerodynamic method of investigation is the fact that by simply varying the number of revolutions of the fan, the head H or Δp can be varied in a simple manner and within a much wider range than in hydraulic test plants. Moreover, compared with water, stable conditions are reached very much sooner when using air, as there are no fluctuations on the surface of the fluid and the masses to be dealt with are much smaller. When using steam as the driving medium, it is difficult to keep both pressure and temperature constant; besides, there is the condensation in the measuring circuits to be considered, which, when neglected, might be the cause of errors.

A NEW GAS TURBINE POWER PLANT

By Prof. Dr. J. Ackeret and Chief-Engineer Dr. C. Keller.

(Condensed from *Escher Wyss Nachrichten*).

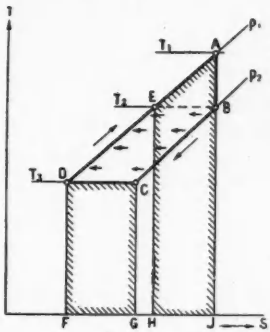
The plant, designed for large output and operating at constant speed, is to be connected with an electric generator. It is claimed further to be capable of operating on various fuels, such as coal, oil or gas, and the thermal efficiency to be at least equal to that of an up-to-date steam plant. The plant operates according to the "constant pressure" principle, i.e., with no periodic inlet or exhaust of gas thus avoiding valves, ignitions and the like. High efficiency should thus be obtained without excessively high pressures or temperatures, and as the exhaust heat is being utilized to the full.

To ensure safety in operation, products of combustion have no access to any moving parts, hereby avoiding losses through friction and wear of the rotating parts.

This new system employed by Escher Wyss, as shown in Fig. 1, is a closed cycle, as compared with the cycle of an ordinary gas turbine shown in Fig. 2. The plant contains a gas heater, a regenerator, a compressor and a turbine. The heat is thus supplied through metal surfaces. The working agent is primarily air, but other gases can also be used.

The process is shown in the entropy diagram, Fig. 3. The expansion in the turbine takes place along the line AB, the regeneration along the line BC or DE. In the compressor the air is compressed isothermatically, whenever possible, from C to D. In actual practice this is done in several stages with intercooling in between. The heat from the outside is supplied to the working medium in the air heater along the line EA.

The area ABCD in Fig. 3 of the ideal cycle represents the work done. It is the difference between the work done by the turbine and that used for the compressor. The work performed by the turbine for 1 kg. is equal to the heat supplied from the outside EAJH. The work used for the compressor is equal to the heat given up to the cooling water of the compressor DCGF.



Entropy diagram of the ideal process operating on the closed cycle system with perfect regeneration.

Fig. 3

Assuming an ideal gas for the working medium, and this is very nearly the case for air, the thermal efficiency of the ideal cycle not considering the losses in the turbine, compressor, gas heater and heat exchanger, and assuming complete regeneration, Fig. 4, is given by:

$$\eta_i = \frac{C_p (T_1 - T_2) - A R T_3 \lg \frac{p_1}{p_2}}{C_p (T_1 - T_2)} =$$

$$= 1 - \frac{A R T_3 \lg \frac{p_1}{p_2}}{C_p T_1 \left[1 - \frac{1}{\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}} \right]}$$

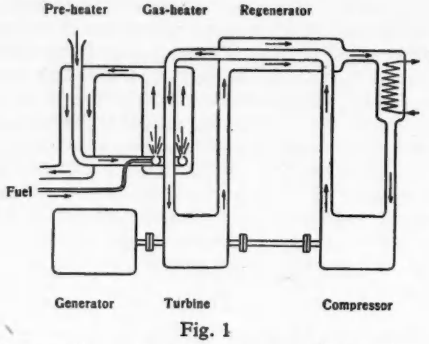


Fig. 1

Diagram of aerodynamic heat power plant working on the closed cycle principle.

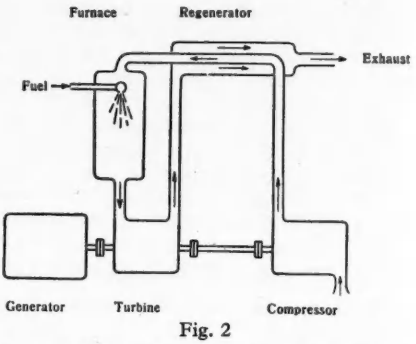


Fig. 2

Diagram of the ordinary open cycle constant pressure gas turbine supplemented by the addition of a regenerator.

This indicates that for a certain maximum temperature T_1 and minimum temperature T_3, the thermal efficiency of an ideal cycle depends only upon the ratio of the pressures. The smaller the ratio of the pressures, the higher the efficiency. This also holds good approximately for a process of operation in which losses occur. Further, for this turbine plant operating on the closed cycle as described, there is no need to increase the pressure in order to achieve high efficiency.

An examination of the entropy diagram in Fig. 3 shows the large amount of heat still contained in the gas, after performing work in the turbine.

This heat is utilized in a regenerator, which heats in counter current the cold air coming from the compressor.

As the entire circuit is made to operate under pressure, the cross sectional areas can be made small, and also the surfaces, for the transmission of heat is nearly proportional to the pressure.

There are other advantages too connected with this pressure-circuit system. The co-efficient of heat transmission on the gas side of the gas heater is better, so that the mean temperature of the wall is lower and is considerably more to the cold gas-side, than it would be the case for low pressure. Both turbine and compressor can be made of small dimensions, and, in spite of this, they operate at higher Reynolds-numbers. Thus with an initial pressure of 140 lbs./sq. in. at the inlet side of the compressor, and for equal temperatures and speeds, the dia-

meters both of turbine and compressor can be reduced to one third. In spite of this, the Reynolds-numbers would correspond to a diameter nine times as great, that is to say, three times as great as compared with a plant of an open circuit. Of equal importance is the ease and convenience with which control can be effected, merely by altering the pressure in the circuit, or in other words, the density, without altering either the temperature or the speed. If the inlet pressure to the compressor were to be reduced to 15 lbs./sq. in., bearing in mind

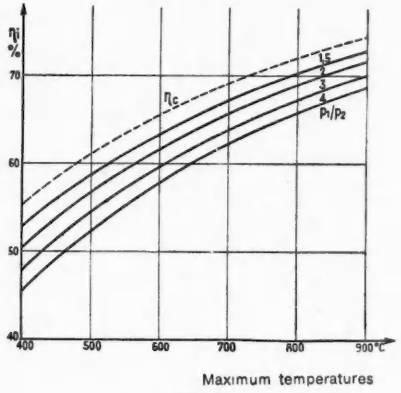


Fig. 4

Thermal efficiency of the ideal process according to Fig. 1 ($T_3 = 300$ deg. abs.)

$$\eta_c = \frac{T_1 - T_3}{T_1} = \text{Efficiency of the Carnot cycle.}$$

the data as stated above, the output would be reduced to one tenth. Since the velocity triangles at all stages of the compressor and of the turbine remain unchanged for constant speed, the efficiencies of the compressor and the turbine are only affected by the Reynolds-numbers. This holds good also for all pressure ratios or number of stages. By lowering the pressure of the closed circuit, and this can be reduced to a vacuum pressure in the closed system, the power to start the plant can be reduced very considerably as well.

The flue gas heat is utilized for pre-heating the air before the combustion, or for some industrial process. The cooled regenerator air is cooled down still further, so that the air at the inlet to the compressor should be as cool as possible. This additional cooler and the cooler for the compressor air are the only sources from which heat is extracted (except the possible losses from the flue gas). It will be evident that the cooling water required for a plant of this kind is only a fraction of that used by a steam plant of the same output, especially as the temperature of the water can be raised considerably without causing any harm.

The total plant actually represents a surface heating plant, not unlike a chemical one. The actual machines incorporated in the plant are of compact design, but the reactions that take place in these machines are of paramount importance for the economical operation of the plant, and fully justify the industrial development and the demand for the most exacting accuracy in the process of manufacture.

Fig. 5 shows an industrial plant of large output, operating essentially as shown on the diagram of Fig. 1.

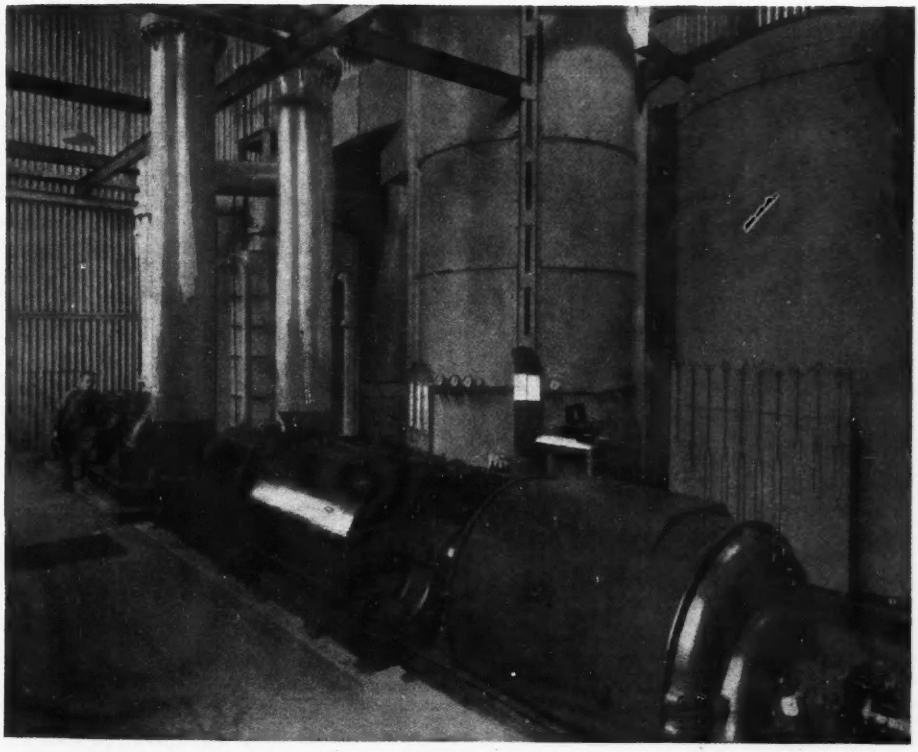


Fig. 5
General view of the aerodynamic heat-power plant of Messrs. Escher Wyss.

STRENGTHENING OF GAS WELDED SEAM BY RECRYSTALLISATION.

By Dr. Ing. Georg Czternasty. (Condensed from *Die Wärme*).

It has been well known that gas welded seams could be sometimes strengthened by hammering them immediately after the welding, or after they have been brought again to red heat. However, the results were rather dissimilar, and the seam was not improved on the average by more than 10%.

This hammering was supposed to densify the seam's loose structure, to improve the granules by destroying the big ones, and to remove slag and gas from the structure. Thus the hammering was performed at the highest possible temperature, immediately after the welding.

Examining this subject more closely it was found that the big granules could not very well have been destroyed, as the seam was plastic at the time of the hammering, and the structure had not been properly cleaned either from slag or gas. But the outer structure was found to be considerably better, especially if heavy hammers had been used. It was found, further, that heat treatment subsequent to deformation was rather harmful. Further, the fine structure of improved seams was found to contain Perlit partially stored in Ferrit, Ferrit granules with and without slip-band formation and marked differences between the granules, all this indicating beginning, or advanced recrystallisation.

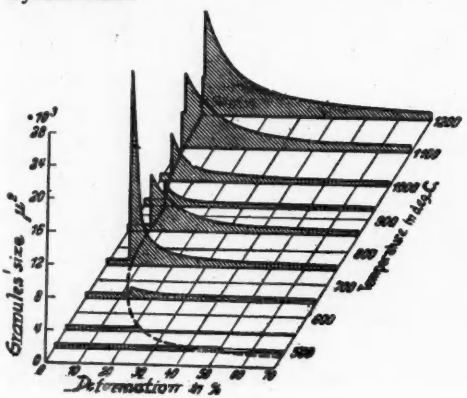


Fig. 1.

Fig. 1 is the well known recrystallisation diagram by *Hanemann* for a steel containing 0.05% C. (similar to that used for soft gas-welding).

It may be mentioned that drawing and compressing effect the recrystallisation in a different way; the latter effecting the crystals to a greater extent. On the other hand it is quite the same whether the deformation is performed before, or simultaneously with the glowing.

The dotted line in Fig. 1 indicates the amount of deformation required at different temperatures to start recrystallisation. The dotted line consists of two marked parts, showing the α and γ phases. This recrystallisation requires a certain amount of time, and if this were shortened by sudden cooling the granules' growth would be checked. Thus there are different ways of getting granules of a certain size, and so the easiest might be selected.

The aim of the improving process is to get the smallest possible granules, and so the temperature ought to be as low as possible, as the speed with which the granules might grow is mostly dependent on the temperature.

Fig. 2 shows the influence of the deformation upon the number of the newly formed granules and that the recrystallisation penetrates the entire structure in 3 to 4 minutes. If it were possible to check the recrystallisation by cooling in the steep parts of the curves, this would result in a particularly great number of granules, i.e., the structure would be very finely

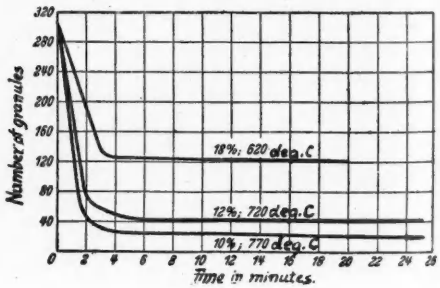


Fig. 2.

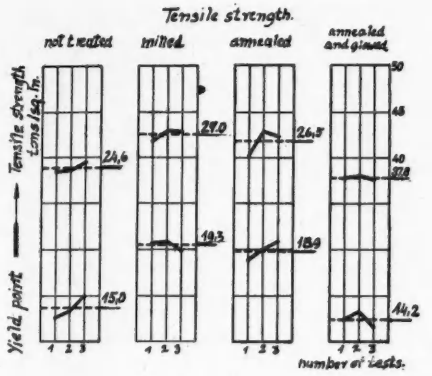


Fig. 3.

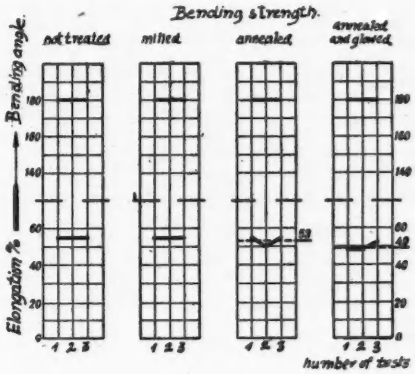


Fig. 4.

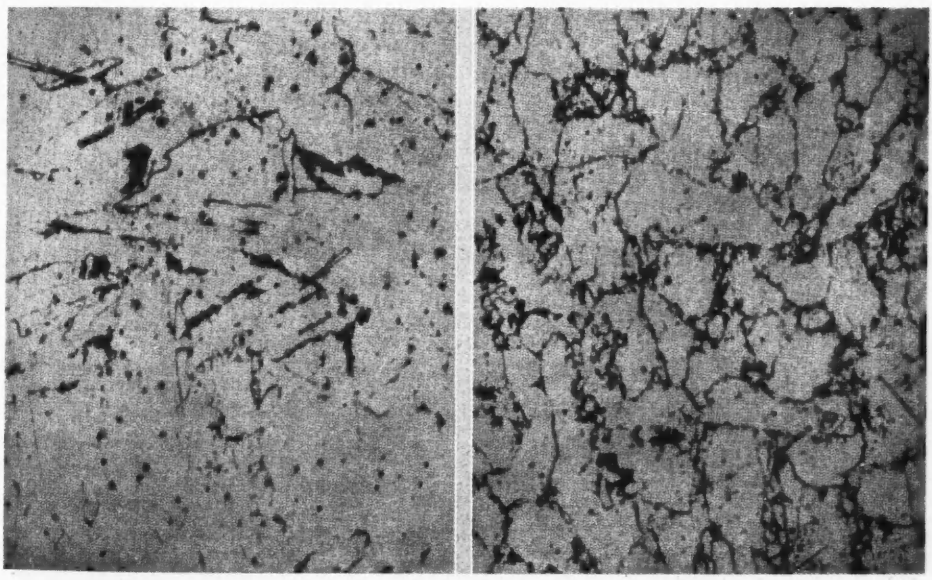
granulated, which could be further improved by bringing it again to red heat.

Besides, *Tamman* found that different impurities passed into solution in the steel separate from the latter during the recrystallisation purifying the seam.

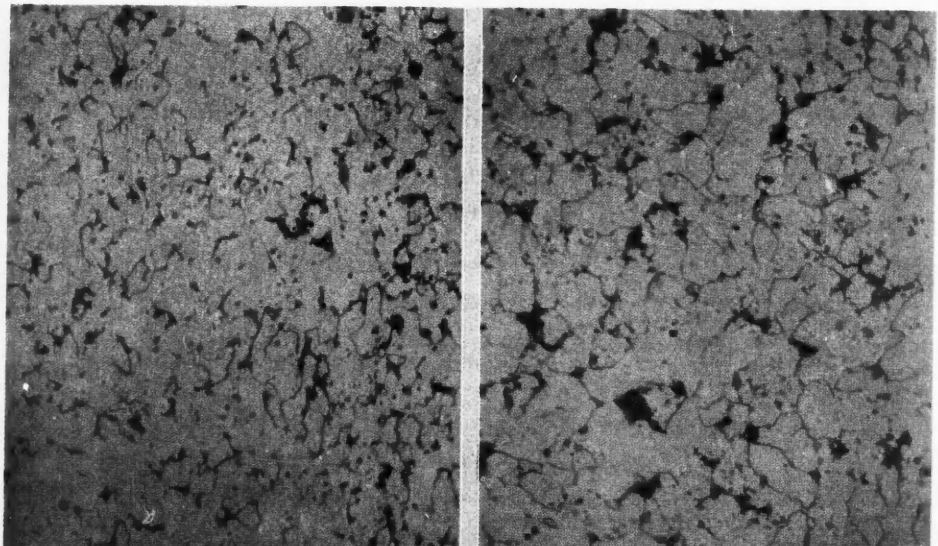
Tests were being carried out with different steel sheets, which received four kinds of treatment :

(1) Usual gas welding, without any further treatment.

(2) The welded seam has been milled at 650 to 700 deg. C., the deformation being 18 to 28%. The seams were afterwards cooled only by air, as only the seam was brought to red heat during the deformation, and it was assumed that the speed with which the cooling

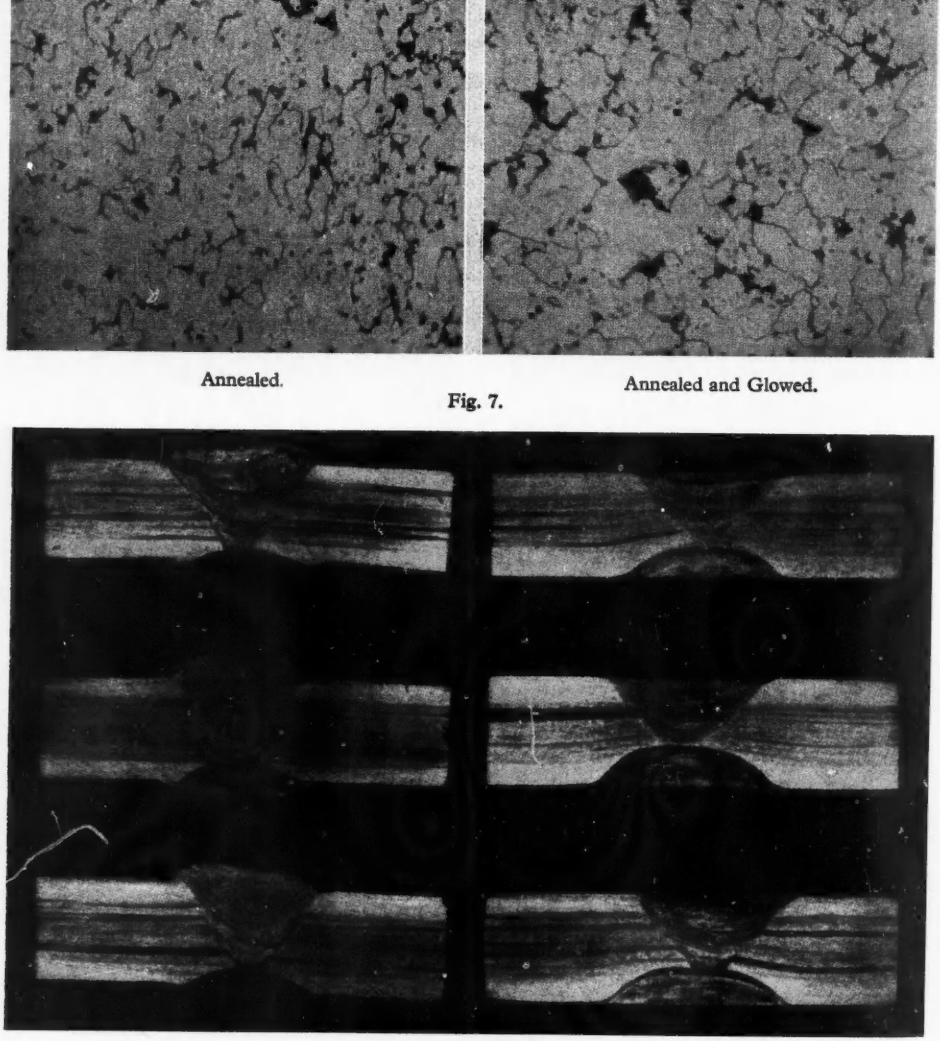


Not treated. Fig. 6. Milled.



Annealed. Annealed and Glowed.

Fig. 7.



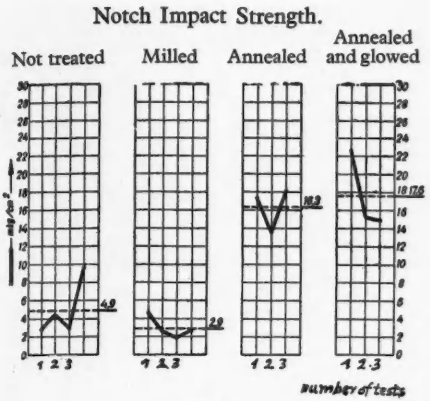


Fig. 5.

took place was sufficient to check the development of big granules.

(3) The same treatment as under 2, but the temperature during the deformation was 700 to 800 deg. C. and it was maintained some time after the deformation had been performed.

(4) The same treatment as under 3, but with subsequent normal glowing treatment.

Figs. 3, 4 and 5 show the result of tests carried out with a steel sheet containing

C 0,11% P 0,024% Yield Point:
Mn 0,43% S 0,036% 22, 2/24, 1 ton/sq. in.

The corresponding structures are illustrated by the photographs in Figs. 6 and 7.

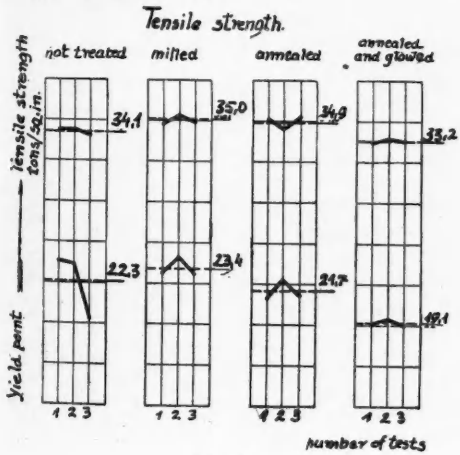


Fig. 8.

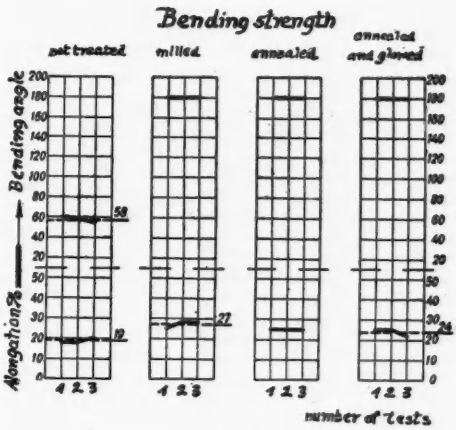


Fig. 9.

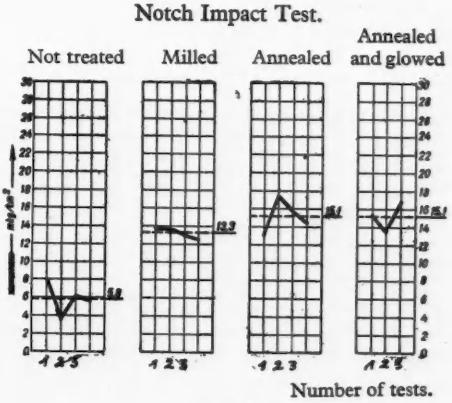
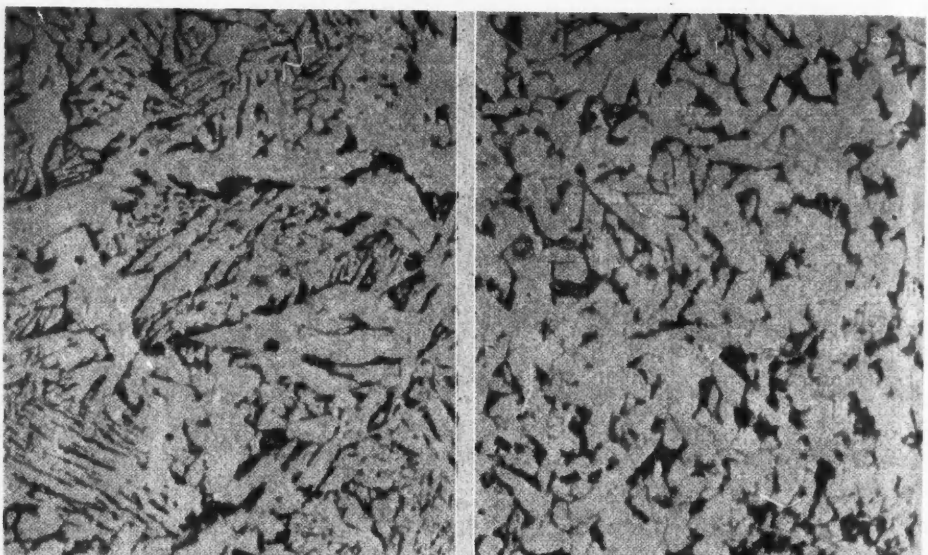


Fig. 10.

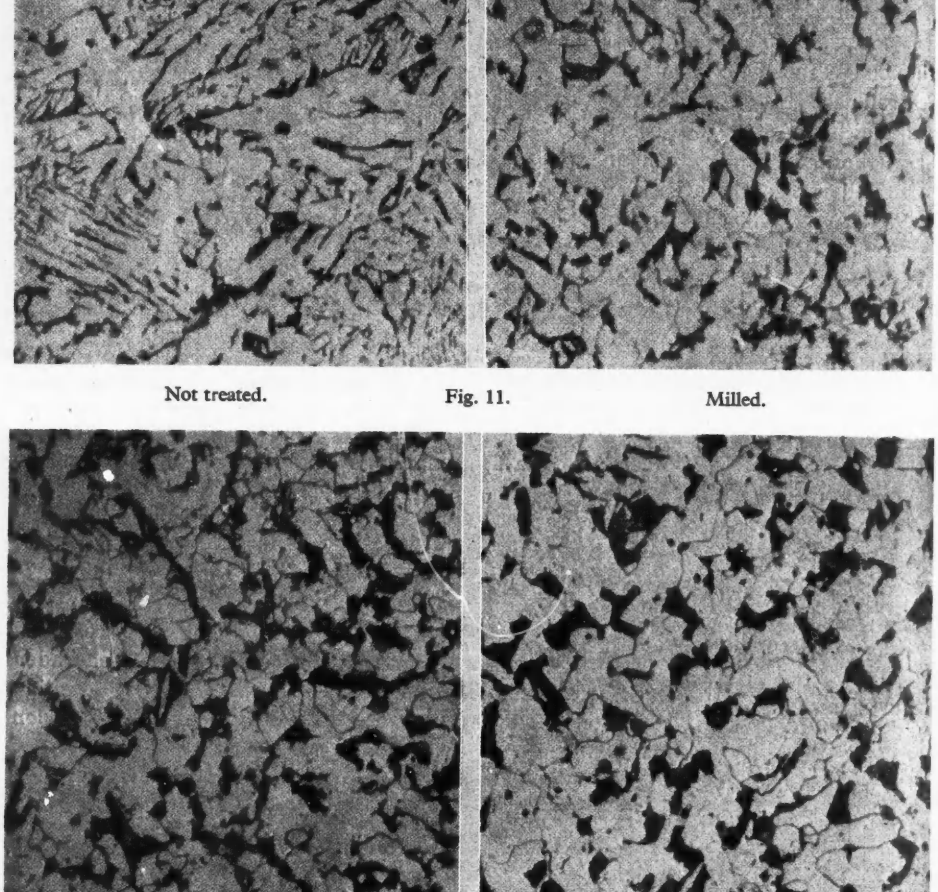
Figs. 8, 9 and 10 show the result of tests carried out with a steel sheet containing

C 0,19% P 0,048% Yield Point:
Mn 0,73% S 0,041% 31¾/32, 4 tons/sq. in.
Si 0,38

The corresponding structures are illustrated by the photographs in Figs. 11 and 12.



Not treated. Fig. 11. Milled.



Not treated. Fig. 12. Milled.

CORROSION OF ALUMINIUM BY ELECTRIC CURRENTS

By E. ZURBRUGG. (Condensed from *Schweizer Archiv für angewandte Wissenschaft und Technik*).

DAMAGE caused by corrosion in connection with electric currents may happen through an enforced outer current, or through small, locally formed elements, the latter being formed by the potential difference of metals in presence of wetness. Thus the nobler parts will become cathodes and the less noble ones anodes, which will be dissolved in due course. So it might happen that on the same surface there will be a great number of cathodes and anodes, and it is rather difficult to measure these local currents with an instrument. Therefore, this case will not be dealt with at this place.

When an ignoble metal is being put into an electrolyte, positive metal-ions will be formed owing to the electrolytic dissolving pressure, which frees electrons simultaneously:

$Me \longrightarrow Me. + \ominus$, these freed electrons giving the metal a negative charge. So the electrostatical attraction between the negative charge on the metal surface and the positive ions compensates the dissolving pressure and results in a balanced condition. Were this balance upset by abducting the electrons, or the ions, a current will be produced and this will start the corrosion. The negative charge may be abducted by putting another metal with smaller solution pressure into the electrolyte and connecting it with the former one. Another way to the same effect would be to connect the metal with the positive pole of a D.C. source, which has its negative pole connected with another metal placed in the electrolyte. Both cases are illustrated in Fig. 1 showing that the current's direction is the same in both cases, but whereas the anode is positively laden in the second case, it is negative in the first. This difference may be of help in tracing the possible source of corrosion: a negative charge of the corroded metal indicating a galvanic element formed by the corroded metal and another one; whereas a positive charge indicates an enforced outer current.

As the amount of corrosion is in direct proportion with the intensity of the current, an open element, although showing a potential difference, does not corrode the anode.

The danger of corrosion for aluminium and its alloys is greater than for the most of the other metals frequently used, as shown in the following register.

Metals	I.	II.
Magnesium ..	abt. −1600	abt. −850
Zink	abt. −1050	abt. −300
Al-Mg-Mn–GA1-Mg-Mn	−760 to −800	−60 to +0
Cadmium ..	−740 to −760	−20 to −10
Pure A1 99, 5% ..	−740 to −750	0
A1-Cu-Ni-GA1-Cu-Ni	abt. −650	+100
GA1-Zn-Cu ..	−850 to −900	−150 to −100
A1-Mg-GA1-Mg ..	−800 to −850	−100 to −50
A1-Mg-Si–GA1-Mg-Si	−740 to −760	−20 to +10
A1-Si-GA1-Si ..	−700 to −730	+10 to +50
A1-Cu-Mg hardened by heat	abt. −650	abt. +100
Tin	−450 to −470	abt. +300
Nickel	abt. −270	abt. +480
Brass 30% Zn ..	abt. −250	abt. +500
Silver	−50 to +50	+700 to +800
Mercury	0	abt. +750
Platinum	+200 to +350	+950 to +1100
Steel	−600 to −700	+50 to +150
Lead	−480 to −500	abt. +250
Brass (50% Zn) ..	abt. −350	abt. +400
Copper	−220 to −250	abt. +500
V2A Steel	abt. + 100	abt. +850

I. being the potential difference against a normal Kalomelectrode in air saturated 2% NaCl solution at 64 to 68 deg. F. in Millivolts. II. being the potential difference against pure A1 99.5 in air saturated 2% NaCl solution at 64 to 68 deg. F. in Millivolts.

As an example the case might be mentioned where the content of an aluminium tank was cooled with aluminium cooling coils, as shown in Fig. 2. The coils soon suffered corrosive damage, and a potential difference of 1000 Millivolts has been found between the tank and the coils. As the coils had a positive charge, this indicated an enforced outer current. The coil S was connected through valves V_1 V_2 with the iron tubes R_1 R_2 to the coolwater pipes W_1 W_2, tubes R passing through the wet concrete floor B. Further, some leaks

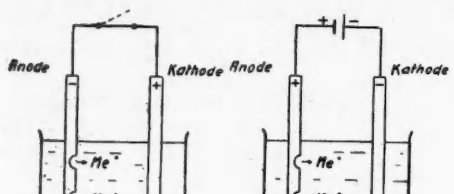


Fig. 1

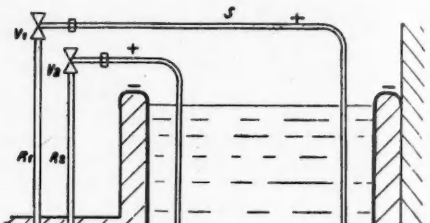


Fig. 2

were found in the tank's insulation through which contact with the wet floor B was possible. Thus a galvanic element was formed between the tank and tubes R and the current was short circuited through the coils and the tank's content.

AUTOGENE SURFACE HARDENING.

By DIPL. ING. THEODOR RIECKEN.

(Condensed from the *Fördertechnik*).

THE far reaching application of autogene surface hardening is suggested for cranes and haulage machinery, particularly as regards cog wheels, worm gears, chain wheels, rotors, axles, gudgeon pins, etc., in order to reduce wear; for this sort of machinery this method is quicker, cheaper and better than others.

Fig. 1 shows the hardness in Brinell numbers as a function of the carbon content

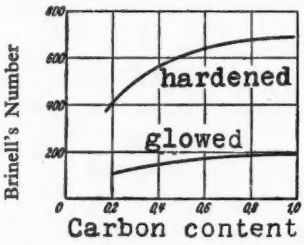


Fig. 1

indicating that the part to be hardened should preferably contain 0.35 to 0.8% carbon.

It is further claimed that chromium-nickel and chromium-molybdenium alloy steels may be hardened as well, provided they contain at least 0.3% carbon. Grey pig iron castings containing 0.5 to 0.8% chemically bound carbon are also suitable.

Coal and acetylene gas are stated to be equally suitable for the process. For the latter a pressure of 17 lbs. sq. in. is recommended, and the oxygen consumption for the former is stated to be about 3.7 times as big as for acetylene.

Special importance is attached to using burners and showers designed for the particular purpose, Fig. 2 showing, for example, the compact device for shafts and axles.

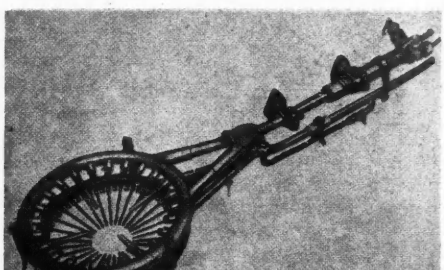


Fig. 2

PREVENTIVE MEASURES TO PROTECT AIR PRE-HEATERS AGAINST CORROSION

By DIPL. ING. ERICH SCHIRM. (Condensed from *Brennstoff und Wärmewirtschaft*).

THE flue gases of a boiler plant are utilised in the last stage only to heat air, and consequently their temperature is not too high at this stage. So if these gases come into contact with the cold parts of the air heater at the cold air inlet, the steam content of the gases may condense. The flue gases might further contain sulphurous acid as well, and this is then absorbed by the condensate, causing serious corrosive damage to the particular part of the air heater, where the condensing occurs.

During the past ten years a number of different designs have been successfully developed to prevent the aforesaid corrosive damage. The basic principle being:

(A) to reduce the corrosion to a certain part of the air heater, which is then designed so that it can be cheaply and quickly replaced.

(B) to prevent condensation by keeping the gas temperature sufficiently high.

(C) to prevent condensation by sufficiently increasing the temperature of the air heater's formerly cold parts.

Fig. 1 shows the embodiment of principle **A**, the replaceable piece being **6**. This arrangement seems to be the simplest of all; however, the boiler's operation has to be interrupted every now and again to exchange part **6**.

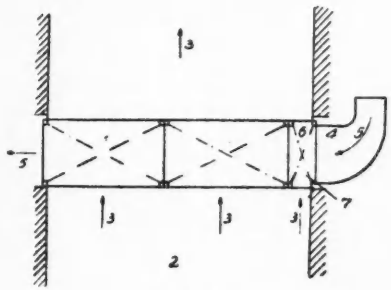


Fig. 1

Fig. 2 shows an embodiment of principle **B**, the flue gases passing first economiser **a** and then air heater **b**, the cold air being supplied by fan **c**. As the right part of the air heater at the cold air inlet, is the part exposed to corrosion, a row of tubes "d" in the economiser below has been removed, and so the gas stream in this part is hot enough not to condense. This design is very simple again however, it involves a little loss in thermal efficiency.

Fig. 3 shows an embodiment of principle **C** by altering the heat transmission's intensity. Three sets of separately controlled shutters **K** may increase the gas stream's velocity at the air

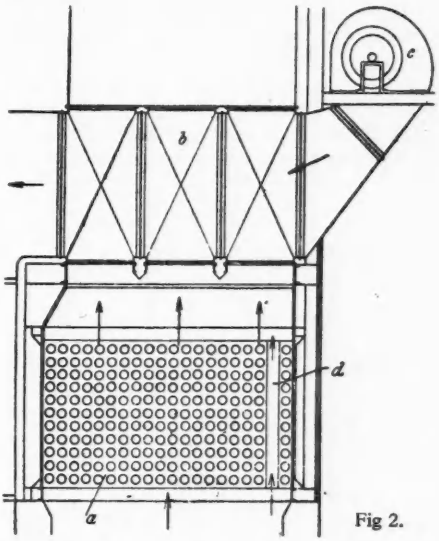


Fig 2.

inlet **a** to such an extent, that owing to the intensified heat transmission, due to the greater speed, the heater's surface temperature rises everywhere above the dew point.

Fig. 4 shows an embodiment of principle **C** by pre-heating the air before the inlet in **k** with steam, or hot water, which are being taken from the upper boiler tank **a** through pipe **f**, or **f1**, and led back to the lower tank **b**. This arrangement has the further advantage that the water, or steam, temperature being always the same and for smaller output less air flowing through **k**—the air temperature increases with decreasing boiler output. The higher air

temperatures thus counter balances the lower gas temperature, due to decreased boiler output.

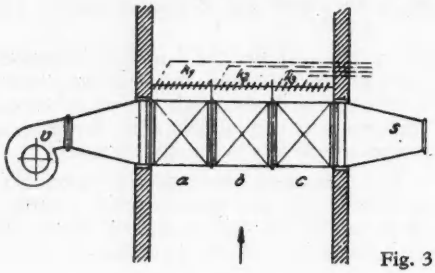


Fig. 3

Fig. 5 shows an embodiment of principle **C** by pre-heating the cold air inlet with hot air taken from the same air heater, **e** being the air inlet, **b** the heating surfaces and **d** the outlet. This design is particularly suitable, if no hot water or steam is available, for instance, in cases when the air heater is not for a boiler.

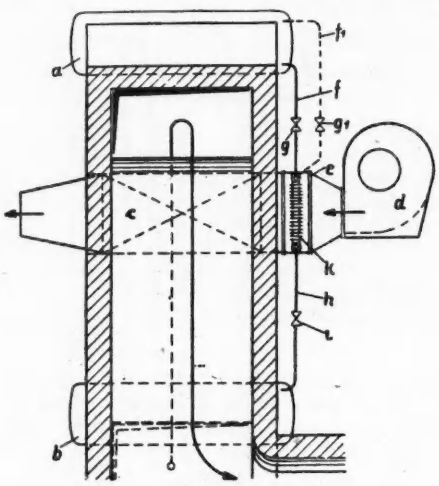


Fig. 4

However, for this arrangement the air should be more intensively heated than usual, the surplus being required for pre-heating the air. The air temperature depends in this case only upon the flue gas temperature, and so this arrangement does not work so favourably as the former one; is, on the other hand, simpler.

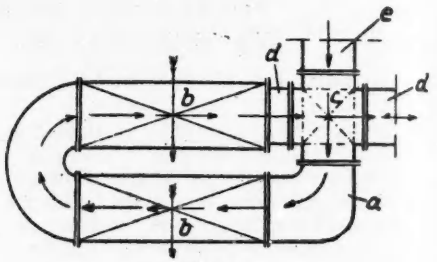


Fig. 5

Fig. 6 is an embodiment of principle **C**, in which the air is conducted at full load through heaters **a** and **b** in counter current to the flue gas, and heater **b** has to be designed so that no condensation should occur at full load, **k** being the air inlet and **h** the outlet. At partial load a certain amount of hot air is being added through pipe **i** to the cold air, thus increasing the surface temperature of heater **b**.

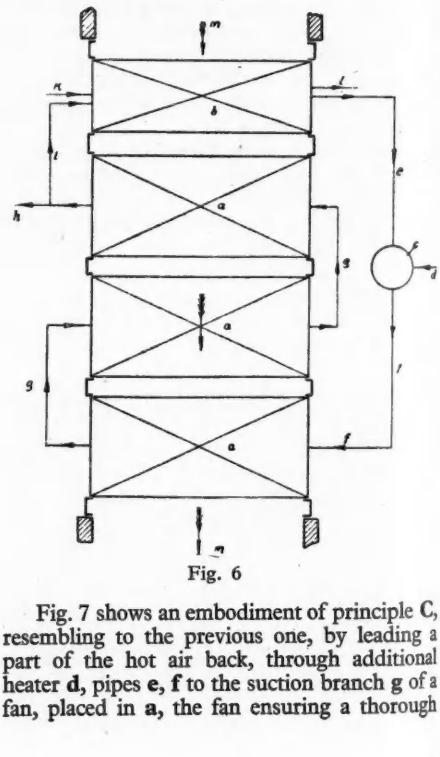


Fig. 6

Fig. 7 shows an embodiment of principle C, resembling to the previous one, by leading a part of the hot air back, through additional heater **d**, pipes **e**, **f** to the suction branch **g** of a fan, placed in **a**, the fan ensuring a thorough

mixture of cold and hot air. As additional hot air's temperature is further raised after main heater **b** by heater **d**, a relatively small amount of air is quite sufficient for safe protection, the governing being in this case rather simple by the means of a flap valve.

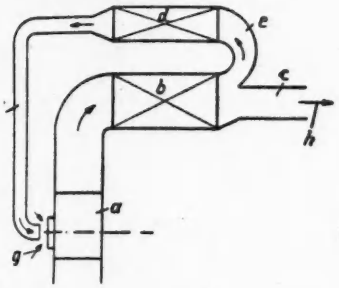


Fig. 7

Fig. 8 is an embodiment of principle **C** by mixing a certain amount of flue gas to the cold air, which raises the latter's temperature, and improves, at partial load, the combustion in the boiler as well. This is further combined with the arrangement, as shown in Fig. 2. Conduit **6** leads a part of the flue gases from before economiser **3**, through governing flap valve **7** to the suction branch **6** of fan **5**, in which the flue gases are mixed with cold air admitted through **4**. Flap valve **7** is further connected through rods **9** with shutters **8**. Thus at full load shutters **8** are open and valve **7** closed. With decreasing boiler output and falling temperature shutters **8** are being gradually closed and flap **7** opened, adding more and more flue gas to the cold air.

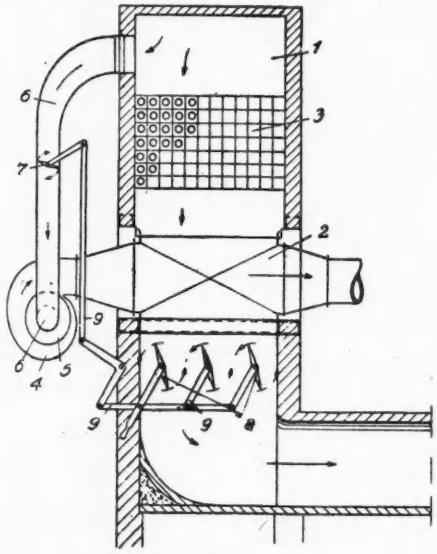


Fig. 8

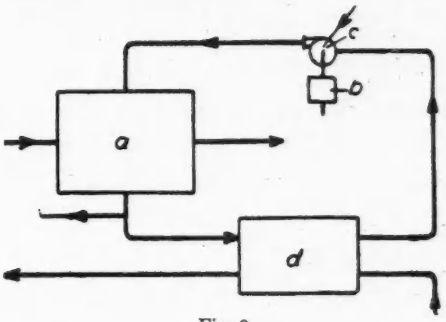


Fig. 9

Fig. 9 shows an arrangement whereby the economiser is being heated by hot air and not by flue gases ; **a** being the air heater, from which a part of the air is being led to economiser **d** and, after passing through the latter, to the suction branch of fan **c** (driven by motor **d**), in which it is mixed with the incoming cold air. This arrangement is stated to have the advantage that eliminates danger of any corrosion in the economiser and thus the economiser may consist of ribbed tubes, saving weight.

Female Labour for Welding.

(Condensed from *Der Betrieb*).

WOMEN were found to be particularly able to carry out quickly and well oxy-acetylene welding of thin sheets. The necessary training for longitudinal and transverse joints of sheets about 5/64 in. thick could be complete in 1½ to 2 weeks with a daily practice of about 8 hours. Occasional help after assignment to actual work was found to increase the output further.

Thin sheets were found to be particularly suitable for training purposes, and the women were found to be rather clever at welding thin sheets of light alloys.

SURFACE HARDENING WITH COAL GAS

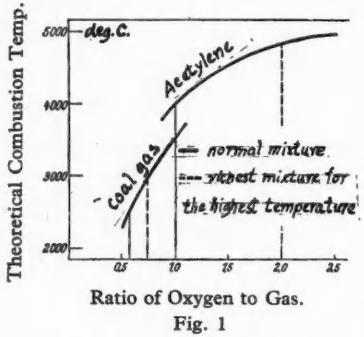
By DIPL. ING. H. W. GRONENGRESS. (Condensed from *Der Betrieb*).

THE process of surface hardening is achieved by the application of heat to the surface to be hardened at a very rapid rate with a view to conduct the least possible heat to the core. Thus the more rapid the addition of heat, the thinner will be the hardened layer. This calls for burners with high efficiency and gases with high calorific values.

Acetylene is superior to any known gases, as regards its calorific value and combustion temperature; however, experiments were carried out to substitute it with the cheaper ordinary coal gas, a comparison between the two showing:

	Acetylene	Coal gas
Calorific value BTU/cu. ft.	1545	481
Max. Velocity of Combustion ft/sec. ..	44′ 4″	23′ 2″
Max. Specific heating BTU/sq. in. sec. ..	6.6	1.86
Max. Combustion Temperature deg. C. ..	3100	2800

On the other hand the temperature of the combustion is greatly effected by the ratio of oxygen to the combustible gas, as shown in Fig. 1, which is calculated and serves only for comparison.



Ratio of Oxygen to Gas.

Fig. 1

The acetylene-oxygen mixture is, as a rule, in a ratio of 1:1, but *Brückner's* experiments showed the highest specific heating performance with a ratio of 2:1. For coal gas the usual ratio used is 0.6:1 and the highest heating performance was found with a ratio of 0.73:1.

In both these cases the highest flame temperature with the highest heating performance is some 20% higher, as compared with the normal one. However, this highest temperature cannot be very well utilized, because it might decarbonise the surface layer.

Subsequent experiments showed further that coal gas can be applied too, and tables 1, 2, 3 and 4 show the practical comparison between acetylene and coal gas.

Table 1.

Rotor of $20\frac{1}{2}$ in. diameter, $1^{37}/_{64}$ in. wide.

	Acetylene	Coal gas
Feed in in. per min.	$13^{25}/_{64}$	$14^{11}/_{64}$
Time for hardening	4 min. 46 sec.	4 min. 33 sec.
Gas used in cu. ft./sq. in.	0.0714	0.118
Oxygen used in cu. ft./sq. in. ..	0.0758	0.077

Table 2.

Surface hardening of a Crankshaft for a 4 Cylinder Engine.

To be hardened: 3 main journals of $2^{2}/_{32}$ in. diameter, $1^{7}/_{32}$ in., $1\frac{1}{2}$ in., and $2^{3}/_{32}$ in. long. 4 crank journals of $1^{13}/_{16}$ in. diameter, $1^{3}/_{16}$ in. long.

	Acetylene			Coal gas		
	Main Journals	Crank Journals	Total	Main Journals	Crank Journals	Total
Time for hardening sec.	35	32		35	32	
Gas used in cu. ft. ..	6,822	7,2	14,02	12,32	13,03	25,35
Oxygen used in cu. ft. ..	6,822	7,2	14,02	6,82	7,2	14,02
Depth of hardening ..	$\frac{1}{8}$ in.	$\frac{1}{8}$ in.		$\frac{1}{8}$ in.	$\frac{1}{8}$ in.	
Hardness ..	58 to 60			58 to 60		

Table 3.

Surface Hardening of Railway Axles. To be hardened: two journals of $2^{3}/_{64}$ in. diameter and $5^{7}/_{8}$ in. long.

	Acetylene	Coal gas
Time for Hardening in sec.	60	50
Gas used in cu. ft. ..	18,75	29,3
Oxygen used in cu. ft.	18,75	13,45
Depth of Hardening in in.	5/32	5/32
Hardness	55	55

Table 4.

Surface Hardening of a Cog Wheel with 21 teeth, $3^5/_{32}$ in. wide with modulus of 10.

	Acetylene	Coal Gas
Gas used in cu. ft. . .	41,3	30,05
Oxygen used in cu. ft.	50,08	18,55
Time of Hardening per tooth . .	143 sec.	35 sec.

The above examples indicate that the time necessary for hardening and the amount of oxygen used are about the same for both gases, whereas to achieve the same effect about 70% more coal gas is required than acetylene. But, whereas acetylene is applied at a pressure of 25,5 to 28,4 lbs/sq. in., coal gas was found to have the best effect when applied with a pressure of 18,5 to 21,4 lbs/sq. in.

It may be pointed out, further, that by concentrating the flame's heating effect with specially designed burners, a sufficiently high rate of heat transference can be achieved for coal gas. This consequence is particularly favourable for big shafts of 20 to 30 in. diameter, as the hardening of these shafts with acetylene is almost impossible on account of the burner's low shifting speed, which is for example for a shaft of only $8\frac{1}{2}$ in. diameter about 59/64 in. per min. On the other hand even for a shaft of about 30 in. diameter the burner's shifting speed is about $2\frac{3}{8}$ to 3 in. per min. for coal gas. This particular fact opened up a new possibility to substitute surface hardened steel rollers for chilled cast ones, the finishing of the former one being considerably easier.

Finally it may be mentioned that no gas producer is necessary for coal gas, as the gas is available in great quantities practically everywhere.

NEW POSSIBILITIES IN HEATING SMELTING FURNACES

By Prof. Dr. Ing. Max Paschke. (Condensed from *Braunkohle*).

The idea is advocated to heat *Siemens-Martin* smelting furnaces directly with gas, if it is available from some bigger producer plant. This is especially advantageous on account of the furnaces' simple construction.

Gas producers for brown coal have been successfully improved lately, and it was found that gas thus produced is particularly suitable for the above purpose, because brown coal gas contains finely distributed coal particles of chemically split tar, which cause the flame to be very luminous. This, in its turn, ensures good heat transmission by radiation.

Gases with high hydrogen content, for example coke gas, have been successfully carburetted with finely powdered brown coal in order to make the flame luminous. The most favourable mixture was found to contain about 0.0125 lbs. powdered brown coal per cu. ft. gas. It may also be mentioned that as powdered brown coal is cheaper per B.T.U. than gas, it reduces the costs.

Possible sulphur content of the gas is eliminated by feeding the producer with some chalk as well.

The design of gas heated furnaces is generally simpler than that of the usual type; and it has been found advantageous to provide ample passages in the regenerative chambers. It is further suggested to design the air preheating regenerative chambers upwardly tapered in order to provide sufficient place for light ashes and slags, and to facilitate their removal. Stones containing more than 50% Al_2O_2 are particularly recommended as building material for these regenerative chambers.

The above-described process will be applied to a 30 t. smelting furnace of novel design, heated at the present with 3 burners, which are supplied with cold coke gas and air of 800 to 900 deg. C. The relatively broad hearth has two operating doors only, and the flue gases are led immediately after the hearth into a small boiler with 184 sq. ft. heating surface, where the gas temperature is utilizedfrom 1700 to 1050 deg. C. The air is pre-heated in continuous counter current by the gases coming from the boiler. This heating system is stated to be 11 to 14% more efficient than the usual *Siemens-Martin* furnaces, because the peak temperature is fully utilised and because the air is preheated in a more economic way.

OIL GLANDS OF BEARINGS

(Condensed from *Technische Rundschau*).

THE American *Garlock* washers are claimed to preserve the elasticity of sealing, to cause very little friction and to resist the deteriorating effects of oil. The washers are stated to be pressed in moulds at high pressure and temperature in V shaped rings, and the material contains neither leather, cork, felt, nor fibre. It is further claimed that the washers are soft and elastic, granular and not porous, have good resistance against oil and low co-efficient of friction.

Figs. 1, 2 and 3 show some interesting constructions in which *Garlock* washers have been embodied.

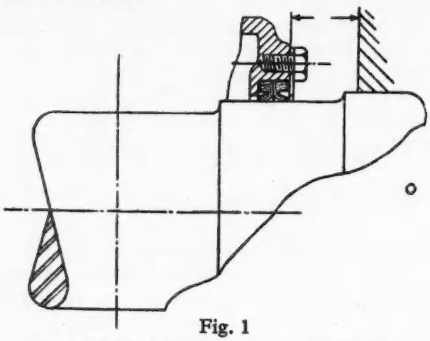


Fig. 1

Double Garlock washer for a railway axle.

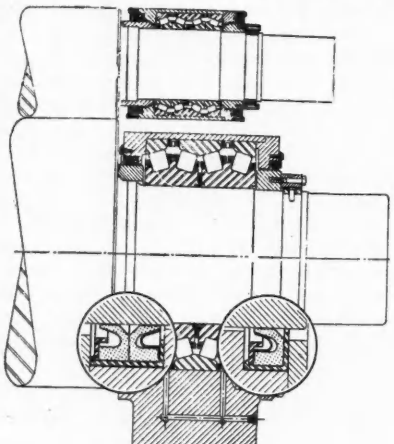


Fig. 2

Garlock washer for the roller of a paper mill.

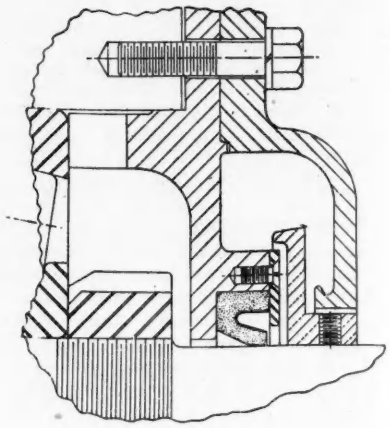


Fig. 3

Garlock washer for the roller of a steelworks.

PREVENTING DAMAGE IN BOILERS

By DIPL. ING. WOELKE.

(Condensed from *Der Maschinenschaden*).

BESIDES incrustation, most of the damage done to boilers is caused by corrosion and by unclean condensate.

Corrosion is caused mostly by oxygen and carbon dioxyde absorbed in the feed from the air. It has been found further that the mere presence of carbon dioxyde stimulates the corrosion by oxygen, the carbon dioxide itself causing corrosion only at high pressures.

As a very simple preventive method it is suggested to keep the water temperature in open reservoirs as high as possible, by heating it with exhaust steam, for as is well known, hot water absorbs less gas than cold water does. Furthermore, the different bicarbonates contained in the water might split up to a certain extent, freeing carbon dioxyde, which thus freely escapes.

Another preventive way is the high alkalinity of the water, which, combined with the former method, is sufficient in most cases.

The above method is, however, not quite satisfactory for high pressure boilers, thermal de-gassing being recommended for this. Alternatively a steady surplus of sodium sulphite in the water is a sure safeguard against oxygen corrosion, but it increases the water's salt content.

THE "GILLI" LOCOMOTIVE

By Dr. ING. GIESL-GIESLINGEN. (Condensed from *L'Energia Termica*).

THE "Gilli" locomotive operates with a steam accumulator, in which steam is stored under an initial pressure of 1700 lbs. per sq. in. as compared with the usual 210 to 360 lbs. per sq. in. hitherto used. Thus the storing capacity of the locomotive has been increased 4 to 6 times.

The basic idea of this design is to store the superheated steam at high pressure and to drive the engine's working cylinders with strongly superheated low pressure steam of 170 to 213 lbs. per sq. in. This operation is illustrated in Fig. 1. The steam enters through valve **a** and filling tube **b**; the high pressure steam **e** passes through outlet tube **d** and reducing valve **f** into superheating coil **g** and finally to the cylinders **h, c** being the water level. The operation of the locomotive is controlled through valve **f**, which is of "Lentz" system and may very well be done by only one man. Further, the filling may happen from a low pressure boiler through intermediate heat pump.

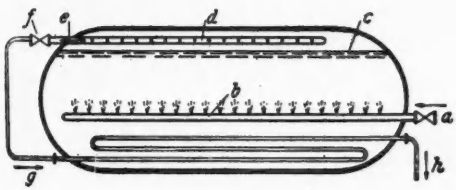


Fig. 1

It may be also mentioned that the storage tank's iron mass helps also to accumulate heat and for the above case it is said to amount to 26%. This is being illustrated in Fig. 2, where the storing capacity is shown as a function of the storage pressure, curve **a** showing the steam storage without reckoning the iron mass of the tank, and curve **b** with the iron mass.

The above system of operation has been embodied in an 82 ton locomotive for a gas factory in **Leopoldau, Vienna,** to transport coal trucks of 1,500 tons into the factory through a gradient of 17%.

Finally a comparison of the "GILLI" locomotive with a usual boiler locomotive

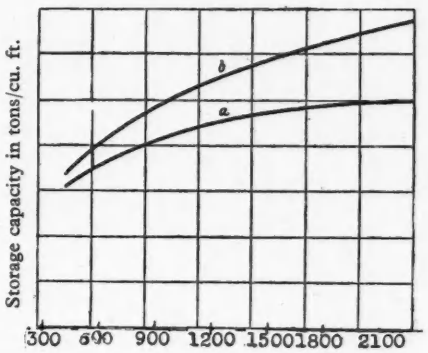


Fig. 2

Initial storage pressure in lbs./sq. in.

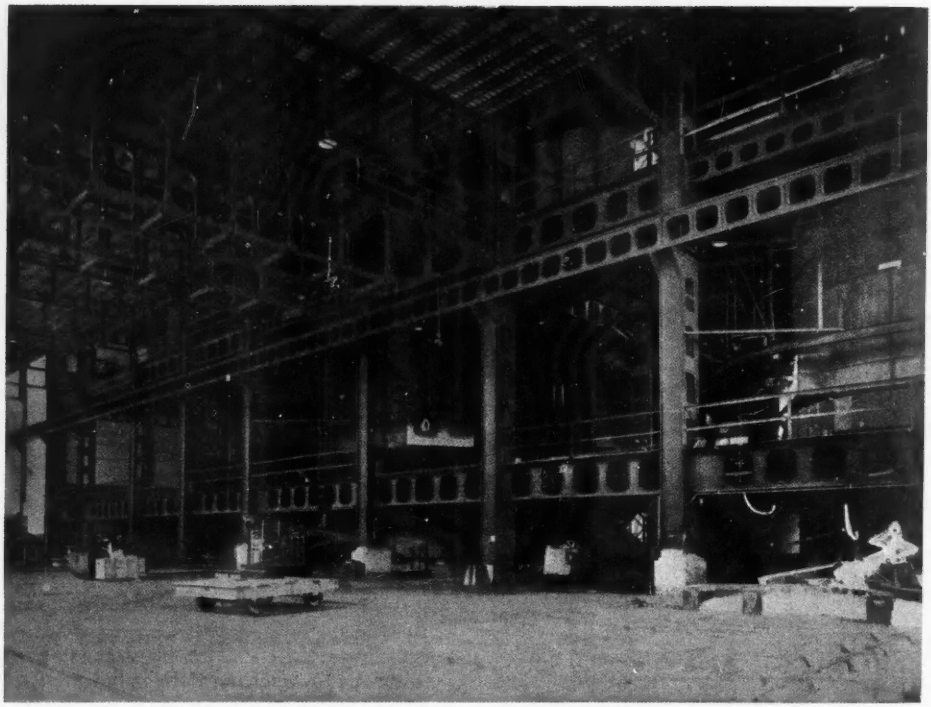
shows that while shunting during approx. 1/3rd of the time coal is being consumed uselessly in the latter, and this figure amounts even to about 40% when not working full days. The corresponding quantity of coal can be economised with the "Gilli" locomotive. In addition to this steam production is considerably more economical with a fixed boiler plant, than in a locomotive and so for the above purpose the savings amount to about 48% in all.

CRACKING OF TUBES OWING TO OSCILLATION

(Condensed from *Der Maschinenschaden*).

THE lower row of inclined tubes in a water tube boiler has often been found to leak. The tubes were about 23 ft. long, with a natural vibration period of 8 to 12 per sec., and the investigation showed that the turbulent gas stream caused these tubes to oscillate.

This led to small local cracks, which then became gradually bigger and caused the leakage.



The three Electric Furnaces in the Ilva Steelworks.

VIERENDEEL GIRDERS IN THE ILVA STEELWORKS AT SAVONA, ITALY.

By A. Bozzarelli. (Condensed from *L'Ossature Métallique.*)

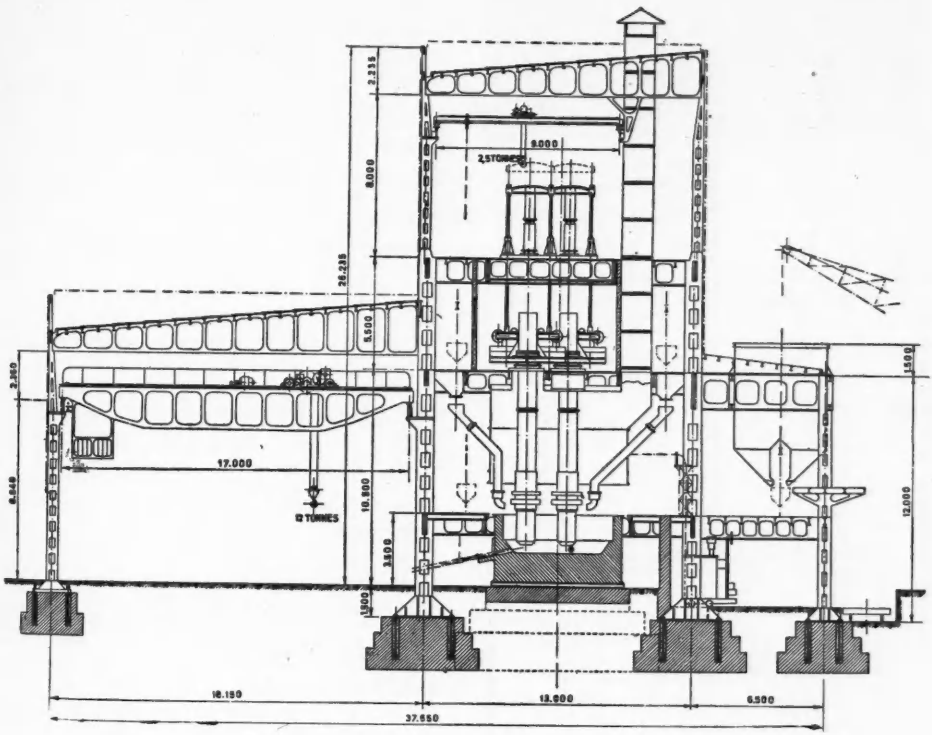
An industrial enterprise cannot be looked upon solely as a source of gain, but must also be regarded as a source of employment and social welfare. One of its aims being that of public utility, it is only right that those who work in the undertaking, spending the greatest part of their day therein, should be accommodated in healthy, well-lit and attractive surroundings. Moreover, even from the utilitarian point-of-view it has been found that the output of workers has shown a remarkable increase in every instance when their working conditions have been improved. Generally speaking the cost of industrial buildings is looked upon as a purely passive form of expenditure which ought to be reduced to a minimum; yet a combination of architectural design, sound construction and careful maintenance cannot but favourably impress the eye as evidence of order and good management. The outward appearance of an industrial building should confer upon it a suggestion of rationality, tying its constituent parts harmoniously together. But if this is to be the theme expressed by the external architecture, neither can the interior be neglected, and the steel framework ought, without sacrificing economy, to be expressed architecturally in lines as harmonious as possible. This raises the question what form of framework is, in fact, best adapted to confer on a structure characteristics of modernity, simplicity and rationality. Thus, framings of

the English, Polonceau or Mansard types are too apt to appear as dense forests of thin lattice-work, lacking in that suggestion of repose which derives from obviousness of structural form and is a characteristic of the plate-webbed girder. In other words, the mechanical impression conveyed to the mind by these constructions does not correspond to immediate simple apprehension of form.

An article in the French Journal *Acier* lays down the following conditions which a steel-framed structure ought to satisfy in order to please the eye : (1) It should express true unity of form. (2) It should make use only of very simple geometrical shapes. (3) The scale of the design should not be excessive ; in other words, the sizes of the panels must not be out of proportion to the depth occupied by the web members.

Large triangulated girders were at one time popular, but are now looked upon with disfavour because of their lack of simplicity. The solid webbed girders which have been much used, particularly in Germany, succeed in suggesting strength and repose, but, however well suited to bridge construction, their look of heaviness is apt to make them seem out of place in building work. The type of girder best of all adapted to these conditions is the Vierendeel, which comes mid-way between the ordinary framed truss and the plate webbed girder.

The general tendency governing steelwork design in recent years has been in the direction of simplicity of line, suggesting modernity, and this is well exemplified in the new works intended for the manufacture of special steels which is named in the title of this article, and



Cross Section through the New Electric Steelworks.

is shown in the illustrations. Among several features worthy of attention are the fact that the component members of the steel framing were assembled by means of welding in the shop and bolting on the site. The girders of the Vierendeel type, are entirely of welded construction, and are supported on steel columns built up from standard rolled sections connected together by welded webs.

The main block which houses the three electric furnaces is a three storey building, 197 ft. long, 64 ft. wide and 86 ft. high, and at the side of this stands a large hall, 243 ft. long by 60 ft. wide, used for the casting operations. The general arrangement will be clear from the illustrations.

Detail of Steel Framed Construction showing the Vierendeel Girders and columns with horizontal diaphragms.

TINDER REMOVING MACHINERY.

(Condensed from *Hydraulik Nachrichten*).

BESIDES a number of different hydraulic sheet drawing and stretching machines, a new process is described to remove tinders from stretched plates.

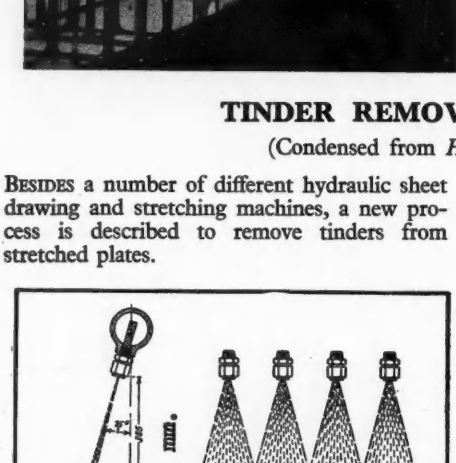


Fig. 1

As shown in Fig. 1 water is projected from specially designed nozzles on both sides of the stretched sheet under a pressure of 850 to 1000 lbs. per sq. in., and the whole sheet passes under this spray. Higher pressures than that indicated above were not found to increase considerably the cleansing effect. The most favourable spray angle was found to be about 15 deg., and the distance of the nozzles from the sheet 12 to 16 in.

In order to use the same apparatus for sheets of different widths the two water distributing tubes are divided inside into four chambers, four nozzles being provided for each chamber and each row of nozzles for a chamber having different widths. Thus, by revolving the two main tubes, sprays of four different widths can be provided.

SMALL IRON-CLAD RECTIFIERS

By E. SCHUPP. (Condensed from *Elektrotechnik und Maschinenbau*).

GLASS rectifiers have been used hitherto mostly for currents up to 500A, on the other hand iron-clad rectifiers have been employed only for currents over 1000A. Thus there was a gap between 500 and 1000A, and as it was found too difficult to build bigger glass rectifiers to fill this gap, the only other solution appeared to be a smaller type of iron-clad rectifier. This has been carried out, and the shape of these smaller iron-clad rectifiers became rather similar to that of a glass rectifier, with distinct anode arms and separated condensing chamber. This design achieved high efficiency regarding the utilization of the vessel's volume, had an almost unlimited life, and is capable of withstanding heavy over-loads.

Messrs. *Siemens-Schuckert* started now to manufacture for 600 to 800A small iron-clad rectifiers according to the above principle. These rectifiers have been used in many plants for speed regulated D.C. motors and are stated to have fulfilled all expectations. These types of rectifiers have been used also to transform D.C. into A.C. The success of these small iron-clad rectifiers was naturally dependent upon the amount of the Voltage drop, as the latter is for bigger glass rectifiers 20 to 22 V. This seems to have been achieved for the small iron-clad rectifiers as well, Fig. 1 showing the drop of Voltage as a function of the current.

An iron-clad rectifier rated for 800A is stated to require the same space as a glass rectifier for 500A. This is claimed to be due to the effective cooling of the vessel's interior. The rectifier has a conical iron vessel and a cover welded to the same, the cathode being arranged at the lowest point and the six anode arms at the side of the vessel. It is further provided with a water cooled cylinder, in which the mercury vapour condenses, the cooling water being re-cooled by air. Thus there is no need for any continuous water supply, as the rectifier works under the same conditions as a glass rectifier, the vessel being cooled by a fan. This advantage has been ensured by keeping the pressure in the anode arms at the lowest possible point. It may be mentioned that the pressure of the mercury

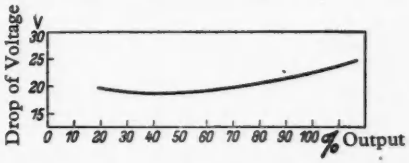


Fig. 1.
The drop of voltage for 800 V

vapour depends upon the temperature in the vessel, and this in its turn upon the amount of the losses. Thus in order to keep the pressure down, the space in front of the anode openings above the cathode has been enlarged, the space above the anode arm openings tapered, and the cross-section of the anode-arm openings increased several times to what was hitherto usual for glass rectifiers. The vessel has been shaped to suit the expansion of the mercury vapour, because the cross-section available for the flow is about proportional to the vapour's volume; this in its turn ensures everywhere in the vessel the right pressure corresponding to the flow. The vessel's shape ensures further the rectifier against reverse ignition, because the anode

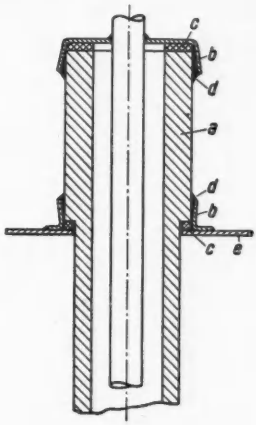


Fig. 2.

a china; *b* cap; *c* support ring; *d* soft soldering; *e* cover.

arms are not situated in the main flux of the vapour. Another advantage is that by cooling the rectifier with air, there is no danger of corrosion, and thin sheets may be used keeping the plant's weight very low. Air cooling being employed the anode arms had to be arranged outside the vessel, as the losses in the same are about as big as in the main condensing space, and the cooled surface had to be increased by fins.

The anode arms can be easily dismantled and the short openings are well machineable. Instead of the usual rubber joints, softly soldered joints were provided permitting the anode temperature to rise even above 248 deg. F. As shown in Fig. 2, extremely simple and long lasting insulators could be used for this design, which is absolutely vacuum tight.

The rectifier with all its accessories is mounted into a relatively very small iron frame as shown in Fig. 3. For the maintenance of vacuum it is further equipped with main pump, precision pump and valve. The accessories are insulated from the frame, whereas the frame is earthed. Safety devices, instruments, etc., are mounted on a nearby separate panel.

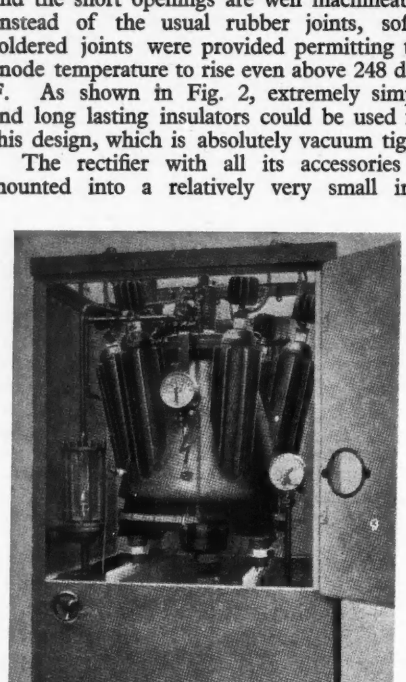


Fig. 3.

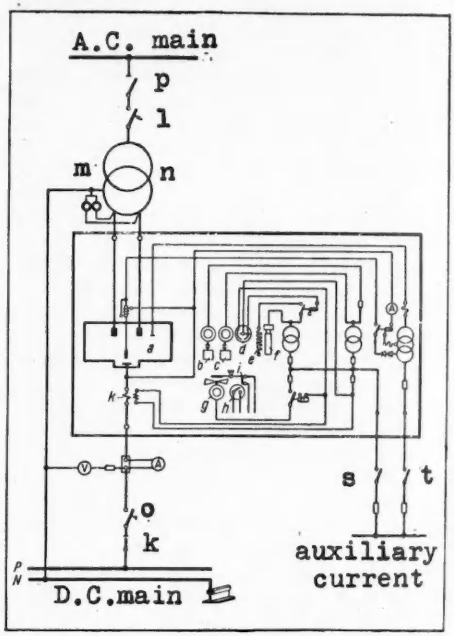


Fig. 4.

a rectifier; *b* water pump; *c* main vacuum pump; *d* thermometer; *e* heating; *f* vacuum precision pump; *g* fan; *h* pressure gauge box; *i* ventilator choke; *k* break switch; *l* protection gear; *m* protection gear against excessive voltage; *n* main transformer; *o* reverse current quick switch; *p* break switch; *s* D.C. auxiliary switch; *t* D.C. main switch.

Fig. 4 shows the diagram of connections. The D.C. auxiliary switch starts the vacuum pumps and the cooling plant, whenever necessary. The D.C. main switch operates simultaneously with the A.C. main switch and starts also the ignition.

The afore-described small iron-clad rectifiers are said to bear overloads rather well and to be particularly suitable for rail service in mines.

METHOD OF EXAMINATION OF THE RELATIVE SWINGING OF OVERHEAD LINE CONDUCTORS

By LADISLAUS VEREBELY. Condensed from *Elektrotechnika.*

FIG. 1 shows a conductor with the sag **FC**, and the maximum angle of its swinging is assumed to be λ. For a fairly approximate calculation it may be assumed further that the swinging follows the sine law. Supposing thus that points **1, 2, 3, . . . , 12** indicate the positions of the conductor in equal time intervals, then the length of the cord is very approximately a function of the sine of the time. Drawing now a line **K-12** perpendicularly to **FC** and a semi-circle with **0-12** as a diameter, the arc of this semi-circle may be divided for example into 12 equal parts. Parallels to **K-12** drawn from the 12 points thus obtained intersect the arc **CK**, and these points of intersection will be reached by the swinging conductor in equal time intervals 1, 2, 3, ... , 12.

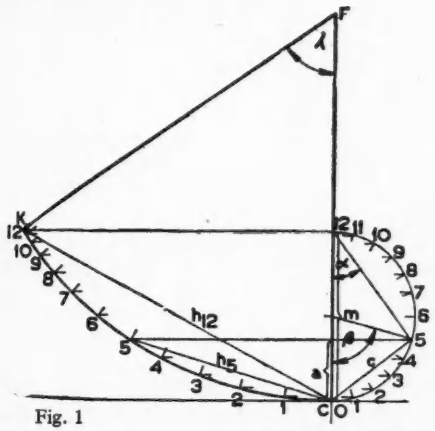


Fig. 1

The above system may be also applied to two, or more swinging conductors to investigate their mutual distances. Fig. 2 shows for instance two conductors with sags **b** and **b'**, their horizontal distance from each other being **t** and their maximum angles of swinging λ and λ'. The points in which the conductors will be at equal time intervals may be now constructed according to Fig. 1, as described above. Points **0-12** and **0'-12'** thus illustrate all possible positions and enable to plot the shortest distance between the two swinging conductors for any difference of phase, 0, 1/12, . . . 12/12. Allowing now **a** to be the shortest distance between the two conductors the inter-sections of circles drawn from points **0, 1, 2, 3, . . . 12** and **0', 1', 2', . . . 12'**, as centres, with radius **a**, show for which phase differences and positions the distance between the two conductors might be less than **a**. This may also be well illustrated by drawing parallel to

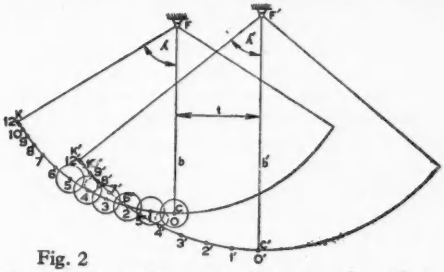


Fig. 2

the abscissa, a line at a distance of **a**, if the distances between 0-12 and 0'-12' are plotted. The points below the line will indicate, too, for which phase differences and positions the distance between the two conductors might be less than **a**.

PHOTOELECTRIC CONTROL IN TRANSPORT AND SUPPLY

By DIPL. ING. FRITZ KOHLER.
Condensed from *Elektrizität im Förderwesen.*

PHOTOELECTRIC cells have been introduced in some places to start escalators working for a limited time when a person passes the control light. This arrangement was found to work best at the time of small traffic and could be switched off during busy periods of traffic. Fig. 1 shows a bunker, into which the supply is photoelectrically controlled.

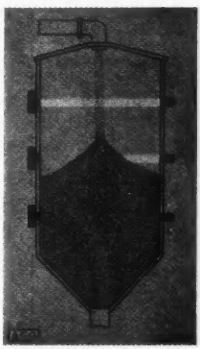


Fig. 1

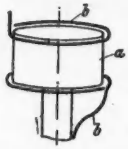
CHROMIUM PLATING OF MACHINE PARTS

(Condensed from *Der Betrieb*).

CHROMIUM-PLATING may be used not only to improve certain machine parts subjected to strong wear, but also to recondition and mend partly worn-out ones.

Suitable for chromium-plating were found to be hardened and unhardened carbon steels, annealed and unannealed nickel-, nickel chrome-, chrome-, and chrome-molybdenium steels as well as grey-cast iron. The nitric layer had to be, however, removed from the surface of nitrated steels before the plating process. The surface should be as smooth as possible, as no chromium layer will form on rough surfaces.

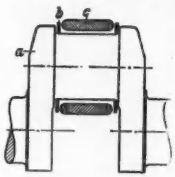
Surfaces not to be chromium-plated should preferably be covered with lacquer paint, and should the chromium-plating process last over an hour, several layers of paint should be applied, or the part should be enveloped with insulating tape and this painted twice.



a piece to be plated.
b wires connected to cathode.

Fig. 1

The layer should be put directly on the base metal at a bath temperature of 120 to 130 deg. F. with a current of 365 to 550 Amp. per sq. ft. and the process should last 1 to 8 hours. It was found to be important to remove the hydrogen developed during the process. On machine parts the chromium layer is usually 1/500 to 1/128 in. thick, the thickness depending upon the allowance for machining. The thicker the layer, the less even it is and the more machining will be necessary. Further it may be mentioned that since the chromium baths do not have a uniform effect, more metal is deposited on corners than on parts lying more inwards. Corners should be for this reason



a crankshaft.
b screen connected to cathode.
c anode.

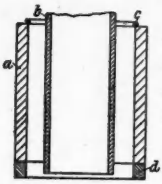
Fig. 2

avoided and rounded off within possible limits; on the other hand this phenemonon may be prevented by properly placed thin sheets, or wires connected to the cathode, as shown in Figs. 1 and 2. At sharp corners well fitting screens, or stencils, are well advised, as shown in Fig. 3.

a piece to be plated.
b screen.

Fig. 3

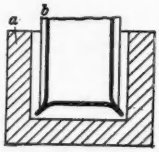
The anodes should be formed to suit the shape. Thus for jobs with holes going right through, hollow cylinders may be used, as illustrated on Fig. 4.; for vessels with corners



a hollow cylinder.
b anode.
c cathode connected wire.
d screen.

Fig. 4

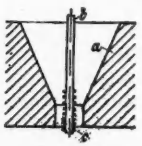
the anode may be well provided with projecting edges, as shown in Fig. 5, to plate the inwardly projected corners; for tapered holes (*e.g.*



a vessel.
b anode with projecting corners.

Fig. 5

nozzles) the anode may be covered with lacquer paint in parts at the narrowest place of the job, as shown in Fig. 6.



a nozzle.
b anode.
c layer of paint.

Fig. 6

Chromium-plating was found to be suitable for all kinds of shafts, guides, cylinders of internal combustion engines, including Diesel engines, presses, pistons, nozzles, reconditioning of worn-out parts, etc.

THE STRESS IN OPENINGS OF BOILER SHELLS

By PROF. DR. ING. E. SIEBEL and DR. ING. S. SCHWAIGER.

(Condensed from *Archiv für Wärmewirtschaft und Dampfkesselwesen*).

LATELY some experiments have been carried out to see the real stress in the openings of boiler shells and to what extent a re-enforcement of the openings would decrease the stress.

For a disc with central bore under tensile stress all round, a re-enforcement alters the situation according to *R. Grammel*:

$$\frac{h}{s} = \frac{\sqrt[4]{k/100 + 1} - 2}{\left[1 - \frac{1}{\left(1 + \frac{b}{r}\right)^2}\right](1 + \mu)}$$

Where

s=thickness of the disc ;
h=thickness of the disc at the re-enforced rim ;
b=width of re-enforcement ;
R=outer radius of the re-enforcement ;
r=inner radius of the re-enforcement ;
σ_0=stress in the not re-enforced part of the disc ;
σ_1=stress in the bore rim ;
$k=(\sigma_1/\sigma_0-1).100$=stress increase in the rim;
μ=*Poisson's* ratio=0.3.

The above formula has been found to apply well enough also to boilers, and the curves in Fig. 1 indicate the required re-enforcement **h**/**s** as a function to its relative width **b**/**r**; the stress increase being 0% for curve **a**, 25% for **b** and 35% for **c**.

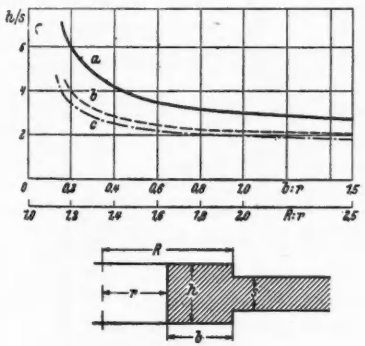
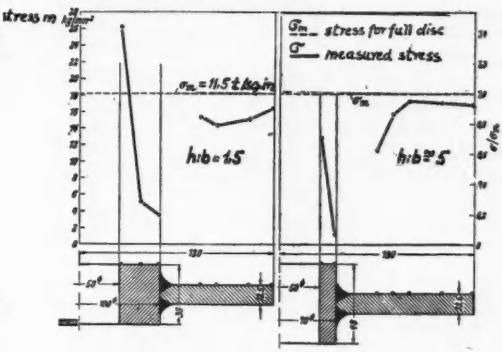


Fig. 1

Fig. 2 shows the results of two tests confirming the aforesaid, σ_m being the stress for the full disc with no opening.



Stress distribution for a load of 5·5 *tons. Measures in mm. ;* σ_m = 11·5 *tons/sq. in.*

Fig. 2

Further tests were being carried out with a tank of $37\,^3/_{16}$ in. diameter, 70 in. long, $^1/_8$ in. thick with openings of 6 to 8 in. diameter. It has been found that the above formula may well be applied, but the actual stress was found to be somewhat lower than calculated, and so it is suggested to allow in the calculation for 25% stress increase. It has been further found that for openings having a diameter of about **d**=**0.3**.D

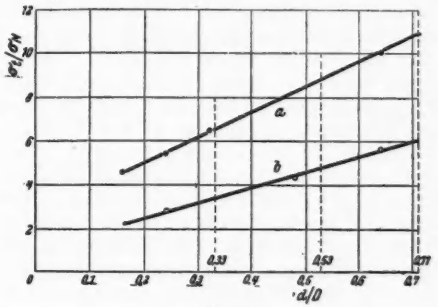


Fig. 3

(**D**=tank diameter) the best re-enforcement is **h/b=3** to **4**. For openings with bigger diameters a re-enforcement of **b=0.4.r** is recommended.

Fig. 3 shows the stress increase at the rim of not re-enforced manholes of diameter **d, D** being the tank diameter; curve **a** being for a wall thickness of $^{11}/_{128}$ in. and **b** for $^{7}/_{32}$ in. Further for **a** it was **s/d=0.9%** and for **b** **s/d=2.2%**, σ_m being the stress for the full drum end.

Fig. 4 shows the stress increase for conical drum ends as a function of the flange angle α, the increase being illustrated, as the multiple of the normal stress $=\frac{1}{2}$ p.D/s. The tests were carried out with drum ends of 10 in. diameter and $^{1}/_{8}$ to $^{9}/_{64}$ in. thick.

The equation of the curves was found to be

$$y = \tfrac{1}{2} + \frac{n}{\tan.\ \alpha}$$

n being for the curves shown 0.8; 1.0; 1.2; 1.4; 1.6 and 1.8.

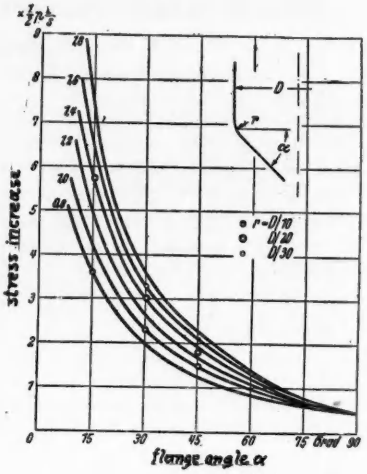


Fig. 4

IMPROVEMENTS IN "BENSON" BOILERS

By Dr. Ing. F. Michel. (Condensed from *Archiv für Wärmewirtschaft und Dampfkesselwesen*).

(1) "Auxiliary Heating Surface."

Boilers with forced circulation have small storage capacity, and with the change of steam consumption the fuelling has to be re-adjusted immediately, it being not sufficient to keep the fuelling at an average level. The different fuelling systems developed lately are elastic enough for this task; there is, however, some difficulty in seeing in time whether the fuelling is really correct. This is so because a change in fuelling manifests itself fully only after a water molecule which, passed into the boiler after the change, leaves the boiler. On the other hand the passage through the entire boiler lasts some minutes and thus some unpleasant wavering might occur.

Messrs. *Siemens-Schuckert* have fitted these boilers with an "auxiliary heating surface" to avoid the above disadvantage. The system is shown diagrammatically in Fig. 1. There is a diaphragm in the feed main and the auxiliary heating surface is connected parallel to it, thus the amount of water passing through the latter will be always in the same proportion to the feed. (The resistance of the diaphragm may be preferably substituted by that of an economiser). This auxiliary heating surface has to be designed so that no steam should

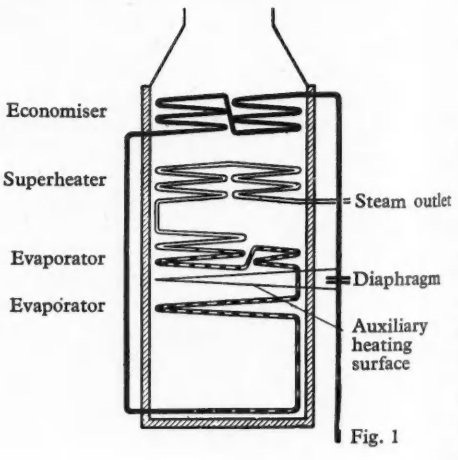


Fig. 1

develop in it, and its surface should be about one-thousandth of the boiler's. Thus the water passes in a few seconds through the auxiliary heating surface, and if the water's temperature is raised always by the same amount, for example 130 deg. F., it is an indication that the evaporation and the fuelling are well balanced. Accordingly the auxiliary heating surface should be preferably placed so as to be independent from the evaporation, in order to simplify the governing mechanism.

In order to facilitate the handling, two pairs of inter-connected thermoelements were used to indicate directly the temperature difference before and after the auxiliary heating surface, as shown in Fig. 2.

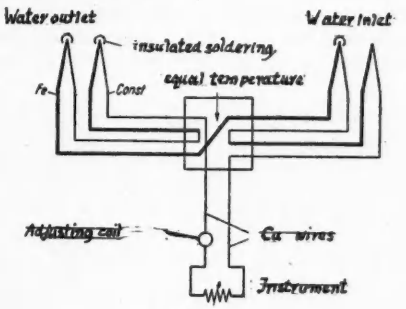


Fig. 2

Good results were also achieved with a remote control mercury differential thermometer, which was provided with specially designed temperature feelers, as shown in Fig. 3. Both capillaries worked upon a single needle pressure difference indicator.

(2) A novel arrangement of Heating Surfaces.

In most cases even the best feed contains some salts, and it is particularily so in ships. As is well known, the greater part of these salts forms a layer in the evaporator of "Benson" boilers, the remainder passing through the boiler in liquid form, or as powder. Accordingly for safety's sake the evaporators were arranged so as to be heated by flue gases of about 800 deg. C., whereas the superheater was heated by gases of about 1250 deg. C. This arrangement worked quite safely, but if the temperature in the superheater suddenly wavered and sank, the salt layer got stirred up and passed into the superheater, where it formed a new layer. This salt layer being at a considerably hotter place caused then serious disturbances.

In view of the aforesaid, the boilers were lately so designed that the evaporator was at a hotter place than the superheater. Thus with sinking temperature in the superheater the stirred salt layer moves harmlessly into the less heated portion superheated. It is even suggested to lower the temperature in the superheater by 50 to 75 deg. F. every other day to remove the salt layer constantly to the less heated superheater.

This arrangement has the disadvantage of necessitating a bigger superheater, but this seems to be compensated by a very safe operation.

Both the aforesaid improvements have been installed in S.S. "Uckermark" and are stated to be working satisfactorily.

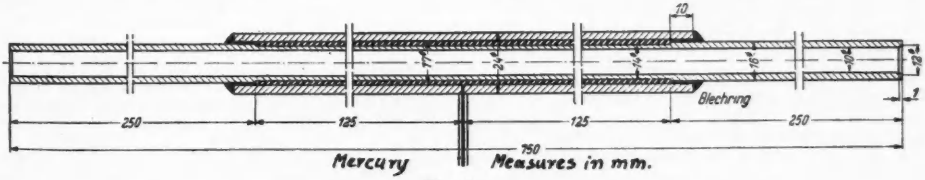


Fig. 3

UTILIZING THE HEAT OF THE EARTH

Condensed from *Sulzer Mitteilungen.*

At the Swiss National Exhibition in Zürich a plant was shown to heat garages, etc., where there is no strict heating, nevertheless the air must not become too cold. The plant comprised an electrically driven fan which drew air from below the earth from a moderate depth. The pump discharged the air into the place to be heated, which thus received also ventilation.

EXPERIMENTAL DETERMINATION OF FLUCTUATING HEAT FLOW

Condensed from *Sulzer Mitteilungen.*

THE fluctuating heat flow through an infinitely large plate perpendicularly to its surface is being investigated by suitably transposing the thermal problem into an electric one. Thus the heat conductivity of the plate is replaced by electric resistance of low capacity and induction, the heat accumulating capacity by condenser and the heat flow by electric current.

Accepting the accuracy obtained through the well known calculation by differences as sufficient, the heat stored in the rod, Fig. 1, between 1 and 3 may be considered, as if concentrated in 2 and replaced by a condenser **C**.

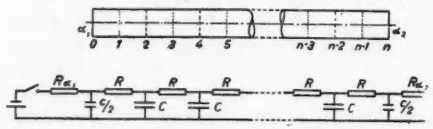


Fig. 1. Electric representation of the non-permanent heat flow in an isolated rod.

The thermal conductivity from 0 to 2 and 2 to 4 is represented by resistance **R**. The capacities of the half end-members 0 to 1 and n-1 to n are replaced by ½**C**. Further the heat transmission to the two ends is represented by separate resistances according to

$$\frac{R_d}{R} = \frac{\lambda}{\alpha \, \Delta \, x}$$

Where

R=Partial ohmic resistance.
R_α=Resistance at the end.
λ=Heat conductivity in BTU/ft/h/deg. F
α=Heat transmission coefficient in BTU/sq. ft./h/deg. F.

Thus a sudden change of temperature at one end of the rod may be represented by suddenly switching on a current from a battery, which has small internal resistance compared with R_α. The electric potential at 0, 2, 4, etc., with respect to the time may be registered by an oscillograph. This, on the other hand, corresponds exactly to the temperature of the plate at the same depth with regard to the time, if

$$R.C. = \frac{\Delta x^2}{\alpha} \cdot 3600 \quad \text{where}$$

$$\alpha = \frac{\lambda}{C_0 . y} = \text{Heat conductivity in sq. ft./sec.}$$

C=Electrical capacity in F
C_0=Thermal capacity in BTU/lbs/deg. F.
=Specific weight in lbs/cu. ft.

In order not to influence the voltage, the ohmic and capacitive resistance both of the oscillograph and the amplifier should be great in proportion to those of the model. D.C. amplifiers with electrometer tubes at the first stage have been found to be suitable.

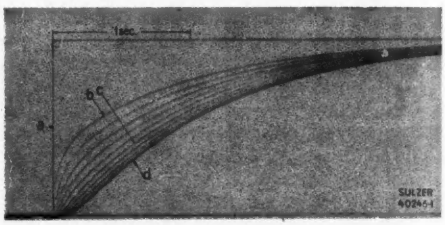


Fig. 2. Temperature increase as a function of time in an iron plate $^{19}/_{128}$ in. thick, when one surface is suddenly heated by steam (α = 8850 BTU/sq. ft./h/deg. C.) and the other side is insulated. *a* Temperature of the steam; *b* Temperature of steam heated surface; *c* Temperature in middle of plate; *d* Temperature of insulated side of plate. The intermediate curves represent the temperatures at different depths of 10, 20 and 30% of plate thickness.

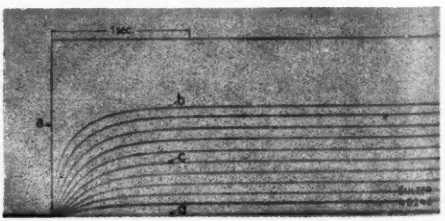


Fig. 3. As in Fig. 2, but with the difference that the side opposite to the steam is always maintained at the initial temperature.

Figs. 2 and 3 show the result of some experiments: the current having been switched on by a contact disc mounted on the film drum. This ensured that the initial points of the time axis exactly coincided. The voltage of the applied current may be varied to represent any periodic change of temperature, as shown in Fig. 4.

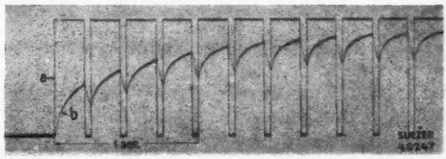


Fig. 4. Sudden starting of periodical heating of a plate similar to that in Fig. 2.
a Temperature of the steam as a function of the time; *b* Temperature at a depth of 10% of plate thickness, as a function of the time.

MINIATURE BALL BEARINGS

(Condensed from *Technische Rundschau*).

THE Swiss firm **Miniaturkugellager A.-G., Biel,** has started to manufacture miniature ball bearings with outside diameters ranging from $^{5}/_{128}$ to $^{51}/_{64}$ in. and in designs, as shown in Figs. 1 to 10.

These miniature ball bearings are intended for the use in cog-wheels, electric motors, registering machines, watches and clocks, automatic weighing machines, speedometers, manometers, various counting mechanisms, etc. Generally they are claimed to be particularly suitable when friction has to be reduced to the minimum and lubrication to be very simple.

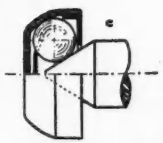


Fig. 1. End thrust bearing with spherical outside bearing surface for taper of 30 deg.

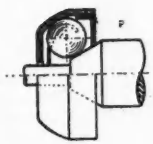


Fig. 2. Collar bearing with spherical outside bearing surface for conic shaft ends, or collars.

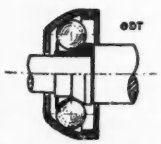


Fig. 3. Collar bearing with spherical outside bearing surface for shafts and spindles.

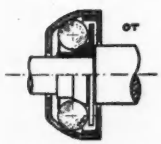


Fig. 4. Collar bearing with spherical outside bearing surface for shafts with end thrust collar

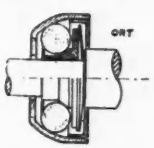


Fig. 5. Collar bearing with spherical outside bearing surface for shafts with end thrust collar & compensating spring.

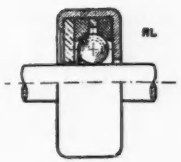


Fig. 6. Radial thrust bearing for shafts moving axially.

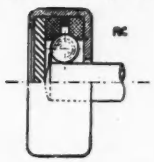


Fig. 7. Radial thrust bearing with supporting plate.

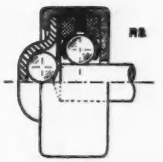


Fig. 8. Radial thrust bearing with end ball bearing to take big radial and end thrusts.

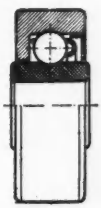


Fig. 9. Radial thrust bearing to take radial thrust as well as light radial loads.

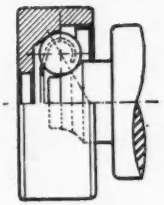


Fig. 10. Massive collar bearing with spherical outside bearing surface; to be used when great thrust and sudden shocks may occur.

MODEL TESTS OF A WIND POWER STATION.

By Prof. J. Ackeret and Ch. Chaille. (Condensed from *Schweizerische Bauzeitung*.)

In the wind tunnel of the Aerodynamical Institute at the Federal Technical University in Zürich the model of a wind power station has been tested. The model was fitted with a variable pitch air screw as shown in Fig. 1, suitable for high speeds, as required for generating electricity.

For a certain output the required wing surface is the smaller, the bigger the co-efficient.

$$\sigma = \frac{\text{Circumference speed of the wing tips}}{\text{Wind velocity}}$$

Thus it seems to be advantageous to use high speed air screws as the entire mechanism becomes smaller.

The efficiency of a wind power station is characterised by the output co-efficient

$$C_1 = \frac{L}{q.F.v} \qquad \text{where}$$

L=Output (ft. lbs/sec)
q=Dynamic pressure=$.^1/_2 \sigma . v^2$ (lbs/sq. ft.)
σ=Air density=γ/g (lbs. sec^2/ft^4.)
v=Wind velocity (ft/sec.)
F=$\frac{1}{4} \pi D^2$ = Wheel surface (ft^2.)

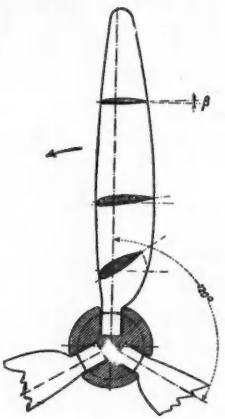


Fig. 1

Fig. 2 illustrates the output co-efficient C_1 for different pitch angles, B=8° and B=20°, and the enveloping curve of all C_1 curves in function of σ, the maximum being C_1=0,44 for σ=5, 3.

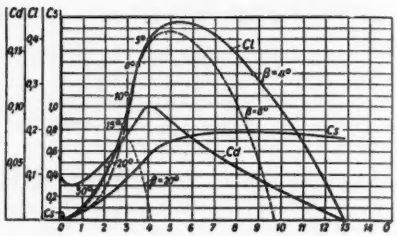


Fig. 2

With a view to these experiments, it has been possible to investigate the control, and it was found possible to keep up a constant speed above a certain wind velocity. At low wind the pitch angles have to be flat, and even storms cannot do any harm if the pitch angle is steep enough.

Fig. 3 illustrates the possibilities of such a wind power station with an air screw of 164 ft. diameter for climatic circumstances, as they are in **Brocken.**

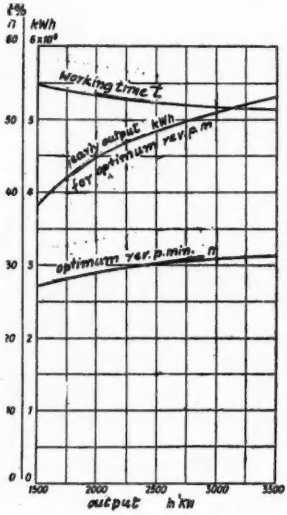


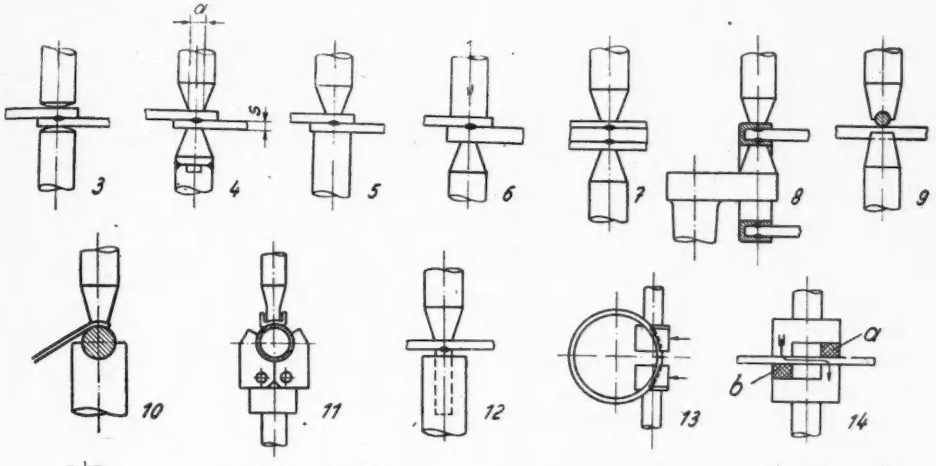
Fig. 3

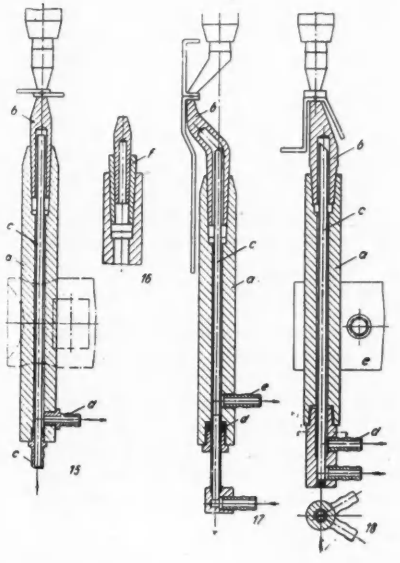
THE DESIGN OF ELECTRODES FOR SPOT WELDING OF IRON AND STEEL PLATES

By OSKAR GONNER. (Condensed from *Werkstatt und Betrieb*).

IT has been found advantageous to design electrodes with a view to the shape of the objects to be welded. Figs. 3 to 14 showing a number of designs most frequently needed. Fig. 4 shows one of the most frequently used designs for which the most favourable dimensions were found to be:

Plate thickness s=	Head diameter d=
5/64 to 1/8 in.	9/32 to 5/16 in.
1/8 to 13/64 in.	5/16 to 15/32 in.
1/64 to 3/64 in.	1/8 to 13/64 in.
3/64 to 1/16 in.	13/64 to 15/64 in.
1/16 to 5/64 in.	15/64 to 9/32 in.





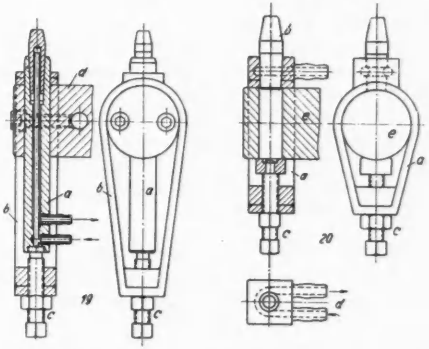


Fig. 14 shows an interesting design also to bring plates up to red heat; *a* and *b* being insulation blocks.

Figs. 15 to 20 show water cooled electrodes with holders, and the smaller and more or less straight ones are fitted with separate short headpieces, which could be used up almost entirely.

AERO-NEWS

(Condensed from *Flugwehr u. Technik*).

New Wing-Profile.

The American Institution for aerial research developed a new wign-profile, which would make speeds in excess of 500 m.p.h. possible. This new profile is stated to cause the air to flow round the wings in contrast to the present pressure waves.

The New " Double-Wasp " Radial Engine.

The new Pratt & Whitney 18-cylinder radial " Double-Wasp " engine's diameter differs only slightly from that of the 14-cylinder engine, and the engine develops 154o B.H.P. at a height of 20,000 ft., which corresponds to an ideal output of 2880 B.H.P. at sea level.

Whereas the usual radial engine weighs about 1.28 lbs. per B.H.P. (at sea-level) and 2 to 2.6 lbs. per B.H.P. with a reduced output at 20,000 ft., the " Double-Wasp " engine weighs only 1.56 lbs. per B.H.P. (at 20,000 ft.)

It is intended to use this engine for flights in the sub-stratosphere, where it would ensure speeds in excess of the usual ones.

A 1450 B.H.P. " Bugatti " Aero Engine.

The well-known French automobile designer, M. Bugatti, has designed a new dobbel 8-cylinder flat lying Boxer engine. The cylinders are of light alloy casting with steel liners, and the camshaft is arranged at the cylinder-heads. The engine is equipped with a supercharger. The reduction gear is equipped to drive one or two air-screws.

The engine's face is supposed to be rather small, so that the engine is very suitable to be built in the wings. The engine develops 1450 B.H.P. at 3000 r.p.m. and its dry-weight is about 1.500 lbs.

A 2900 B.H.P. Aero Engine.

The " American Aviation " reports that an aero engine with 2900 B.H.P. output has been designed in Michigan. The engine has 24 cylinders arranged in H formation and is made entirely of steel, no aluminium being used. The engine weighs 2780 lbs. and its petrol consumption is 0.25 lbs. per B.H.P.h. at 2160 B.H.P.

An interesting feature of the design is the arrangement of an air-screw at the fore and the aft of the engine, mounted so that the engine's longitudinal axis is perpendicular to that of the aeroplane.

A New 4000 B.H.P. Aero Engine.

The Wright Works in the U.S. have started to build a new type of engine with 4000 B.H.P. output. The engine is said to be built up of six 7-cylinder radial engines, mounted behind one another.

New Blind-landing System.

A new blind-landing apparatus has been tested in Lakehurst, U.S. Two cables were laid in the ground through which A.C. of 500 cycles was passed. There were two entirely separate aerials in the aeroplane, wound round the aeroplane's body. When the current is switched on in the cables in the ground, each cable influences another aerial. There was a Voltmeter attached to each aerial to indicate the induction current.

Supposing the aeroplane is flying at a height, which is equal to half the distance of the two cables in the ground, the instruments will indicate a certain current. Further the two cables are laid down so as to converge to a point. Thus if the instruments show steadily the same current, the aeroplane descends at the required space, being always at a height which is equal to half the distance of the two cables below. A sideward deviation is indicated when the two Voltmeters show different currents.

The entire receiver set is stated to weigh only about 39 lbs. and to cost very little.

As a further advantage it was mentioned that the cables may be laid down in hilly terrain, but in this case the height difference of the ground has to be taken into account by altering the distance between the two cables accordingly.

New Blind Landing Apparatus.

The American Wireless Corporation has patented a new blind landing system, for which no transmitter is necessary on the ground. The apparatus consists of a short-wave transmitter, mounted in the very end of a wing, and of a receiver set, tuned to the above sender and set in the very end of the

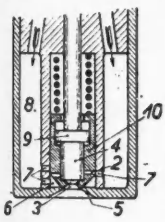
other wing. The sender's beam is directed towards the ground and the beam will be received in the receiver after having been reflected by the ground. If the landing place is surrounded with a metal band, the reflection of the beam will be considerably stronger than at other places. Thus the pilot sees from the receiver set's instruments if he is above the landing place. Further, the same apparatus indicates the height as well according to the well-known Echo principle.

PATENTS AND INVENTIONS

Oil Burner suitable for a Wide Range of Supply.

(Condensed from the German Patent No. 682763, owned by BROWN, BOVERI & CIE, Baden.)

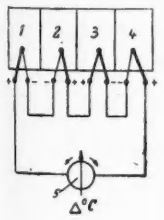
THE adapting of the injector burner to different charges is being performed by governing the oil inlet openings. As shown in the figure below, the nozzles **3** and **5** are co-axial, and part **4** of spindle **9**, while moving upwards, frees first inlets **2**, to which the oil gets through openings **7**, forming at the same time an oil ring in **6**. By the further upward movement of spindle **9** the piston-like inner nozzle **10** is being subsequently lifted, thus opening nozzle **5**. The different oil inlets, nozzles and openings are arranged with a view to ensure a quite even and smooth change in supply.



Electric Control of the Operation of Internal Combustion Engines.

(Condensed from the German Patent No. 680375, owned by SIEMENS & HALSKE A.-G.)

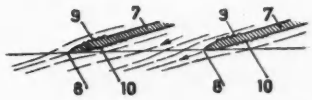
SERIES connected thermoelements are arranged in each cylinder of the engine, as shown in the figure below, and these elements are further connected with an instrument, which indicates a possible temperature difference in the cylinders. The instrument may be connected also separately with each thermoelement to find out in which cylinder the disorder occurred.



Guide Blades for the Low Pressure Part of Steam and Gas Turbines.

(Condensed from the German Patent No. 681479, owned by ESCHER WYSS MASCHINENFABRIK, G.m.b.H., in Ravensburg.)

TESTS carried out on the boundary layer bordering of guide blades showed that the alternating sphere of the turbulence depends mostly upon state of the same, and consequently upon the **Reynold's** number. Experiments showed also that guide blades, which had a rough surface near their tips only, caused the flow to keep near their surface, and this being so even in the sphere of small **Reynold's** numbers.



Automatic Petrol Governing.

The Pellegrini - Bugiardini automatic petrol dosing governor, lately developed in Italy, is stated to save on the average about 20% petrol.

The mechanism adjusts the petrol inlet to the engine's momentary output. This is achieved by a petrol tank fitted with a cone, which reduces, or enlarges the petrol inlet, according to the engine's output. This is done by a pressure gauge box, connected to the suction branch and a cone in the gauged carburettor outlet. Increasing vacuum in the engine's suction branch thus reduces the jet's transverse section and consequently the petrol supply.

Treatment of Piston Rings.

The outer faces of piston rings are lately coated with tin copper cadmium, etc., in order to shorten the running-in period. Good results are now claimed with phosphate coatings. After a short while a blank layer is stated to be forming both on the piston rings and the cylinder. Additional graphite treatment is stated to improve further the phosphate coating.

Flexible Shaft Spring-Coupling.

Condensed from the German Patent No. 687037; owned by ROBERT DOME and PAUL JAROSS, Budapest.

As shown in Fig. 1 the two journal ends **1** and **5**, fixed to spring **3**, are supported each in a bearing, mounted in **4** and **6**. Bearing supporters **4** and **6** are coupled with each other through joints **8**, which are situated so that their axis **x-x** passes through point **o** where the axis of the bent shafts **1** and **2** intersect each other.

This arrangement is said to be particularly suitable for perpendicularly swinging system of shafts, substituting cardan shafts. It is claimed as an advantage that the torque transmitted through this coupling is quite even and independent of the angle at which the coupled shafts are to each other.

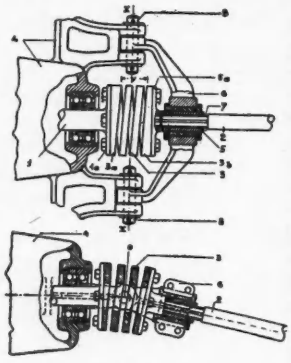


Fig. 1

Air, or Oil Pump for Autocars.

Condensed from the German Patent No. 687169, owned by DAIMLER-BENZ A.-G. Stuttgart-Untertürkheim.

As shown in Fig. 1 several valve stampers of the engine are constructed as pump pistons. Thus stamper **3** actuated by cam **1**, sucks in air, or oil, through opening **5** and delivers it under pressure to pipe **9**, through back pressure valve **8**. It is also claimed that under certain conditions piston rod **10** might be dispensed with, and compressing space **4** might serve as an elastic cushion for the valve gear.

This pump is recommended particularly as a substitute for a compressor providing compressed air, or oil, for actuating the brakes.

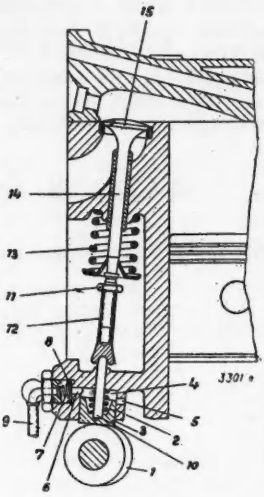


Fig. 1

HIGH SPEED ROTARY PUMPS

(Condensed from *L'Ingegnere*).

IT is being pointed out that high speed rotary pumps, as shown in Figs. 1 and 2 have been lately built to deliver ¼ to 400 tons per hour of liquids. The efficiency of these pumps is stated to be about 80% and to be almost independent output. As a particular advantage, it is claimed that these pumps are lately designed for special purposes so as to be able to supply fluid in both directions without any loss of efficiency. This is carried out by building the left side of the pump, shown in Fig. 1, in the same way, as the right side.

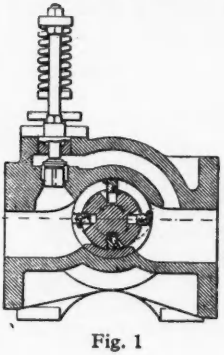


Fig. 1

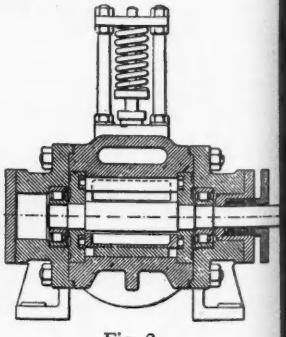


Fig. 2